MARLIN-ROCKWELL CORPORATION
M-R-C Research and Development Laboratories
Jamestown, New York

N64-29508
Code-1-CAT-17

[redacted]

Final Report

JET PROPULSION LABORATORY
California Institute of Technology
Pasadena, California

Survey of Frictional Problems in Spacecraft

Purchase Order No. N2-150168

February 16, 1962

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FOREWORD

A survey of frictional problems in spacecraft mechanisms has been conducted in order to determine the degree of research and development effort which is required to solve the problems introduced by space environment. The survey was conducted in three parts:

A. A visit to Jet Propulsion Laboratory, Pasadena, California, for a period of one month to review design layouts and drawings of proposed spacecraft mechanisms and to discuss problem areas with cognizant JPL personnel.

B. Visits were made to personnel engaged in research and development activity related to problems associated with space environments.

C. A survey has been made of existing literature to review the present state of the art.

This report covers all phases of this program including recommendations for further action, and is submitted in fulfillment of the requirements of the subject contract.

We wish to acknowledge the important contribution to this report made by Dr. Wilfred E. Campbell whose dissertation on electrical contacts appears in the Appendix. Several members of Marlin-Rockwell Corporation Research Staff, especially J. H. Gustafson and G. A. Bentley who were essential to its preparation, and Mrs. Mary Rosen and Mrs. Lois Hammond, to whom fell the task of typing the manuscript.

SUMMARY

During residency at JPL, it became apparent that frictional problems may be grouped according to similarities and that mechanisms and space experiments which appear to be extremely diverse actually share identical design problems. It is also apparent that research and development effort applied to antifriction bearings may be of equal value in the areas of gear design, electrical contacts, and plane bearings. As a consequence, these areas are included within the scope of this report.

Before beginning this survey, we believed that much could be found in the literature that could be applied to problems in space environment. For example, we felt there would be results of testing in inert gas environment that would apply to the problems associated with vacuum. We finally concluded, however, that such is not the case. On the contrary, much of the data which has been generated in vacuum equipment and which appears to bear directly on the problems of space environment is of little value.

Recommendations are made, recognizing the pressures of the time schedule for space exploration. Initial recommendations are our best guesses, based on past experience, for use in craft of the immediate future. Recommendations for subsequent research and development are made to produce maximum results in a minimum length of time in keeping with the increased complexity of future missions.

It is concluded that research work directed to frictional problems in bearings, gears, seals and electrical contacts is required within the scope of JPL activity. Research should concentrate on implementing design decisions, developing new mechanical design concepts and basic investigations to develop materials and lubricants for the future. More emphasis on lubrication is warranted.

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REPORT OF VISIT TO JET PROPULSION LABORATORY

1. SCOPE OF JPL ACTIVITY

JPL has three major programs of which two, the Lunar program and Planetary-Interplanetary program, involve frictional problems in a space environment. The Lunar program consists of three projects: Ranger, Surveyor and Prospector.

Early Ranger tests were scheduled to be orbital flights in a high elliptical orbit near escape velocity. (The second Ranger shot, November 18, 1961, was, like the first, only partially successful since a low earth orbit was achieved.) These experiments are being conducted to test the control system, power supplies, communication equipment, and determine the life time of components in space. Later tests will include impact landings on the moon.

Following the Ranger series, Surveyor spacecraft will be sent on soft landings on the moon. The contract for design and development of Surveyor has been awarded to Hughes Aircraft. While the Ranger Lunar landing package will contain seismological instruments, the Surveyor package, in addition, will include instruments to measure the moon's gravity and a drill that will bring a specimen back into the craft for chemical analysis. Later Surveyor missions may include small roving vehicles, such as those described in Aviation Week (Ref. 1).

The Prospector spacecraft is still in the planning stage and is intended to be launched by a Saturn rocket. It will be designed to rove over the moon's surface. It will perform numerous experiments in final preparation for placing a man on the moon.

The Planetary-Interplanetary program originally consisted of the Mariner A spacecraft, intended to fly by, orbit and land instruments on Venus, and the Mariner B, a similar craft to be launched to explore Mars. Delays in the development of the Centaur boost vehicle have caused a compromise of the Mariner A program. A lighter weight, modified Ranger spacecraft will be used for Venus exploration. A larger craft, called Voyager, is in the planning stage for interplanetary work (Ref. 2).

2. JPL ORGANIZATION

Organizationally, the Jet Propulsion Laboratory consists of nine separate divisions of which the following four are primarily concerned with frictional problems in space environment:

a. Engineering Mechanics Division

b. Propulsion Division

c. Division of the Space Sciences

d. Guidance and Control Division

The Engineering Mechanics Division is responsible for the spacecraft structural design and for mechanical integration of other subsystems. This effort includes most of the elements of a complete system which are required to accomplish the mission, such as: spacecraft bus, landing craft, exploration devices. This group is also concerned with research and development in the fields of materials, structures, dynamics, fluid mechanics, thermodynamics, mechanisms, and environmental survival equipment. Fortunate the board design group is largely concentrated in this Division, permitting close coordination of design solutions to friction problems.

The Propulsion Division is responsible for propulsion systems which are to be used in the spacecraft. This includes thrust vector and impulse control, retro rockets for landing, propulsion devices for surface exploration and return flight power. The large booster rockets for initial take-off are not part of JPL responsibility.

The Guidance and Control Division is responsible for navigation equipment and auxiliary power sources such as batteries and fuel cells in solar power turbine generators. They are also responsible for attitude control actuation.

The Division of Space Sciences is the group responsible for instrumentation and experimentation aboard the spacecraft. No two instrument packages are identical and, consequently, this group is working on an extremely large group of instruments.

As a result of discussions with various members of these groups, the following applications are presented as being typical of spacecraft mechanisms.

3. SPECIFIC APPLICATIONS

3.1 Gear Boxes:

The gear box which unfolds the antenna on the Ranger vehicle is shown on Drawing J-3151748 (Ref. 16). This box is powered by a Kearfott 5000 RPM motor. On Ranger 1 and 2 the gear reduction is 234,000 to 1. On subsequent vehicles the reduction is 78,000 to 1 and this same gear box may be employed on Mariner vehicles. Gears are lubricated with Electrofilm 4396 molydisulfide type lubricant and bearings are lubricated with a thin film of GE 300 grease. In general, specifications call out shielded bearings and phenolic retainers impregnated with GE F50 oil. These are obtained if they are available and can meet delivery. The operating life requirement is 100 hours.

The Mariner gear box is shown on Drawing J-4200007 - Universal Actuator Assembly (Ref. 32). This gear box is driven by a Curtiss Wright stepping motor which has an output torque that varies from 8 oz. in. at the start of a pulse to 2 oz. in. at the end of the pulse. The motor operates at 25 pulses per second and provides

36 degrees of revolution per pulse. The output of this gear box is 10 ft. lbs. The reduction is 1800 to 1 so that output is two hundredths of a degree per pulse. In the motors as received, the star wheels are made of nylon. These are replaced by anodized aluminum star wheels. There are ratchets inside and out and these have give little or no trouble. The life requirement is 1000 hours on this gear box. This may possibly be reduced to 500 hours.

Both of these gear boxes are pressurized with either nitrogen or dry air. Both boxes are sterilized at 257°F for 24 hours. The Ranger box is also sterilized with ethylen oxide gas.

The lives given above are operating lives including both preliminary test, ground check-out and mission operation. The over-all shelf life requirement is approximately two years. For example, all Ranger parts have been ordered for nine Ranger missions. The ninth will probably occur two years after the parts have been ordered.

The Ranger gear box has a worm drive which operates in conjunction with a helical gear. They have had trouble with molybdenum disulfide dry film as a lubricant on this pair of gears which they feel is due to extreme sliding forces. The Mariner box contains all spur gears and so far, has not given any difficulty.

A question has been raised relative to the torsional stiffness of bearings. Torsiona stiffness is a problem in these gear boxes due to the large overhanging moment of the antenna. On the Mariner vehicle they are considering a change over to a United Shoe Machinery Harmonic Drive, since it gives them greater torsional stiffness. That driv has a full complement of uniform size balls for the wave generator. The torsional stiffness required in the Mariner gear box is 7500 lbs. ft. per radian.

Both of these gear boxes are servo type actuators and after they have unfolded the antenna, they dither constantly to keep it pointed in the right direction.

Future trends with respect to these gear boxes are anticipated to be:

a. There will be more of them on each vehicle.

b. They will become more complicated.

c. They will not be able to be pressurized and will have to operate in the vacuum of space. Life requirements will be extended as interplanetary flights become longer and interplanetary landing operations become extended. In some instances they are talking five years continuous operation at 100% reliability.

d. Torque requirements and loads will tend to increase as more complex machinery comes into use.

3.2 <u>Lyman-Alpha Scanning Telescope:</u>

The following description of this experiment aboard Ranger 1 and 2 is given to indicate the types of motion and friction involved (Ref. 4).

The telescope weighs about 14 pounds. The assembly is mounted in a gimbal system which permits motion of the telescope axis in two perpendicular planes. Elevation drive is effected by a synchronous motor with the direction of drive determined by 2-element commutators. The upper contact reverses the direction of motion while the lower contact initiates the azimuth motor to step out 2° in the azimuthal plane. On the completion of this azimuthal motion, the elevation motor is permitted to drive again, the azimuth remaining unchanged, while another line is scanned. After completion of this scanning operation, the telescope remains in its final position until the next picture is to be taken. Fifteen pictures are scheduled at distances from 8.9 to 183 earth radii.

The elevation motor is mounted on the gimbal ring and drives through a worm gear to actuate the telescope. The azimuth motor is a stepping motor directly driving the gimbal ring. At the opposite end of the gimbal ring axis is the azimuth potentiometer which is gear driven from the gimbal axis by a single set of spur

3.3 Ranger Solar Panels:

The solar panel actuation system on Ranger 1 and 2 is composed of an actuator, two hinges, and four latches on each panel. In the folded position the panel is supported on the two hinges and the four latches. Panel opening is initiated by a signal which triggers the squib power supply. The power supply fires the explosive cartridges at each latch and the latch pins are retracted. Then the panel is free to move and the actuator pushes it into the open position and locks it.

The time required for the panel to open to full position is a function of both temperature and load. In a zero G field the opening time varies from 120 seconds at 20°F to about 33 seconds at 120°F. In a one G field at 80°F, the opening time is about 12 seconds, whereas in a zero G field at 80°F the opening time is 45 seconds. Silicone oil is used for the damping fluid in the actuator.

The actuator characteristics are as follows:

Weight	-	$16\frac{1}{2}$ ozs.
Output (retracted)	-	150 lbs.
Output (extended)	-	30 lbs.
Over-all length retracted	-	10 in.
Stoke	-	3 3/8 in.

During initial tests, long and erratic operating times were experienced. This was caused by high friction loads between the barrel and the guide and was solved by coating the barrel of the actuator with molybdenum disulfide dry film.

The solar panels are latched with explosively actuated pins. The tongue is fastened to the solar panel and the other components are mounted on the spacecraft structure. The pin passes thru a hole in the tongue to hold the tongue to the forward structure.

3.4 Ultraviolet Spectrometer Assembly:

This will be used on a trip to Venus. The ultraviolet spectrometer has a diffraction grating which is required to rotate through an arc of $20\frac{1}{2}°$. The grating is mounted on two ball bearings with a 3/8 in. bore. A lever overhung on one side of the shaft follows a cam, using a ball bearing as a cam follower. No lubricant coating can be tolerated on the optics.

The entire unit sits idle during the trip to Venus. Then it will scan the surface of Venus during flight past the planet.

Power in this unit is supplied from an American Electric variable reluctance stepper motor No. 11S12J-15D. This motor has an angular step of 15°. The cam is driven from the motor through two stages of spur gearing. The cam and its pinion are mounted on two bearings of 3/8 in. bore. The first stage pinion is mounted on two 1/4 in. bore bearings. The cam runs at 1 RPM.

The gears are as follows:

> Motor pinion - 10 teeth, 96 pitch, .1042 pitch diameter. Its mate has 150 teeth, 96 pitch, 1.5625 pitch diameter. The other pinion has 20 teeth, 80 pitch, .250 pitch diameter. Its mate has 240 teeth, 80 pitch, 3 inch pitch diameter. In this gearing, back lash is not especially critical. As this unit is now designed, the gears and bearings are sealed from the optics with O-rings. Gears and bearings will be lubricated by molybdenum disulfide and F50 oil with GE 300 grease respectively, as in the gear boxes described previously.

This spectrometer is similar to the Lyman-Alpha telescope application previously described and is presented as another example of a recurring type of design.

3.5 Horizon Sensor:

The Horizon Sensor is shown on Drawing No. 4253-001A (Ref. 24).

This unit has been subcontracted for design and engineering to the Barnes Engineerir Company, Stamford, Connecticut. It will be used on the Mariner which means that it will sit idle for three to six months during the voyage to Venus and will have a mission operating life of 10 to 20 hours to scan the Venus planet. However, severa hundred hours running life are needed on bearings to permit ground check-out.

The main telescopic unit is supported on three Kaydon thin section, ball bearings. The largest has approximately $5\frac{1}{2}$ in. bore and the other two bearings, 4 in. bore.

The two 4 in. bore bearings each support a prism and can be rotated relative to eacl other for different light incidence angles. These two bearings and their housing a supported in the single large bearing which supports the entire telescopic unit. Torque is extremely critical in this application and is the reason for using only one rather than two support bearings.

One of the 4 in. bore bearings turns at 315 RPM, the other at 105 RPM. The outer support bearing rotates at a rate of 1 to 2 RPM but has a maximum total travel of 90°. This is ±45° oscillation. These bearings have stainless steel races and are four point bearings. The retainers are Nylon (may be impregnated) and the balls wi be coated with OBS coating. This unit will be pressurized with dry nitrogen and helium.

3.6 Lubrication of Explorer III Cosmic Ray Tape Recorder:

The Explorer III satellite was launched on March 26, 1958. Cosmic ray data were re- ceived from the satellite until June 5, 1958 when the transmitter went dead. Althoi there were some interruptions in telemetry during this period, it is believed that tape recorder was operable during the entire 72 days of the satellite's active life

The tape recorder was not sealed and was, therefore, operated in space in vacuum ranging from 10^{-6} mm Hg at the 121 mile perigee, to approximately 10^{-12} mm Hg at the 1746 mile apogee. The temperature inside the pay load was measured to range between 0°C and 35°C. This is probably about the temperature environment of the recorder.

In the record mode, the unit was driven by a stepping motor which, through a gear train, advanced the tape past the recording head at a rate of .005 inch per second. This resulted in rotational speeds of 4 revolutions per hour to 100 revs. per hour for the various gear stages. Once during each orbit the recorder was triggered on command and played back the information gathered. Since the period of the orbit was 115 minutes, it is assumed that this is the average time for a record cycle. The play back cycle took place in 5 seconds, after which the recor cycles was automatically resumed. During play back, the tape is driven by a sprin motor, which was wound during the record cycle. In this mode, rotational speeds ranged from 96 RPM to 2400 RPM for the slowest and fastest components respectively

All rotating members were mounted on miniature ball bearings (some shielded), even the cam followers used for switching. Bearings were lubricated with a di-ester oi conforming to the Specification MIL-L-6085A. The exact oil used is believed to have been Winsor Lube L245X, but this has not been completely confirmed. All gear were operated dry.

The bearings used in this recorder are as follows:

a. NHB - FFR-144PP
b. MPB - SS4FM
c. NHB - FR-144PP
d. MPB - SS5P
e. MPB - SS-2.5FCHH
f. MPB - SS-4-FM

Bearing housings were magnesium and shafts were of stainless steel.

The gears were all made of stainless steel and had a diametral pitch of 64, and a pressure angle of 20°. The number of teeth varied from 12 in the small pinion to 45 in the largest gear. All gears were pressed on the shafts and the shafts mounted in miniature bearings.

3.7 Mariner Boom:

The Mariner boom is typical of a short life-short storage unit (See Section A, 5). It consists of three telescoping tubes. The particular one that I saw had one section of black anodized aluminum, but the other two sections were treated with MoS_2 because the anodizing was not sufficiently dark. The darkness is required for temperature control. Each section is lubricated by GE 300 grease which is contained in a small pocket formed by two red silicone (silastic) O-rings at the end of the tube immediately outside. Test stand actuation of this device showed that the lubricant left a very sticky coating upon the tubes. It is believed that the large surface area absorbs the oil in the grease leaving the thickener to smear out on the tube surface as a sticky residue. Additional tests will be run after the tubes have been wet down with F50 oil and the grease thinned out by addition of more oil. This sticky residue was primarily apparent on the black anodized surface. The MoS_2 surfaces felt slipperier.

A similar boom is used for the Ranger RA 3, 4 and 5 gamma ray experiment. It will mount on top of the space frame hex and extend, following the mid-course maneuver.

The boom consists of three sliding tubes and one stationary tube mounted concentrically and to retain internal pressure. A latch mechanism retains the boom in the retracted configuration and utilizes a pin puller identical to those on the solar panels. The central tube serves as a pressure chamber and is charged with dry nitroge which forces the boom to the extended position when the latch is released. A special double O-ring grease packing is employed on all sliding seals. The extended position will be maintained by internal pressure, eliminating the need for mechanical locks.

The boom extends 6 ft. from the fully retracted to the extended position. Its retracted length is approximately 30 inches. Diameter of the boom is $2\frac{1}{2}$ in. and the weight about 5 lbs. (Ref. 5).

3.8 Magnetic Drum:

An interesting unit is being designed which requires placing a message on a magnetic drum which will be read by a tape head in space to instruct the mechanism for its duty cycle. The magnetic drum is to be an aluminum cylinder coated with ferrous oxide. The problem is how to lubricate the ferrous oxide coating to prevent seizure between the drum and the tape head.

One suggestion is to m x the ferrous oxide with molydisulfide in a phenolic resin binder and coat the aluminum cylinder in one operation.

Other suggestions were to glue Mylar tape, in which the ferrous oxide was embedded, to the surface of the cylinder. Another suggestion was to apply a Teflon coating as lubricant over the ferrous oxide.

3.9 Manipulator:

General Mills has just delivered an experimental manipulator, built under contract, for research and development for possible future space applications. This is designed to handle a television camera on the moon. It has three degrees of freedom. First, it will oscillate about an inch from side to side, it will rotate about its vertical axis $\pm 180°$, it will elevate about a horizontal axis $\pm 180°$. Motion about each of these axes is derived from a specially designed motor by Globe Motors Company. It is a hysteresis type motor, 10,000 RPM, specially built with Teflon winding to withstand 400°F temperature (max.). All bearings in the motor and gear boxes are Bartemp bearings. The gears are coated with Teflon.

These drives are otherwise similar to other drives which have been described previously. There is, however, an interesting difference in this device. Motion about the vertical axis is with outer race rotation. This places the bearing between the gear box seal and the outer environment. Consequently, as the spacecraft would gain altitude and move into areas of reduced pressure, there should be a rush of air from this chamber to the outside through the bearing.

There is a chain drive which connects the horizontal axis drive motor with the gimbal ring, housing the Lunar television camera. This is the only instance, in any JPL mechanisms, of the use of roller chains. It is not known whether any attempt has been made to lubricate the chain or the sprockets.

3.10 Lunar Soil Sampler:

The Lunar drill (Surveyor) is an impact, star type drill powered by a rotary impacter in which the pneumatic cushion is replaced by a resonant spring. The drill rotates at 2000 RPM.

The pulverizer is a particle collision type, rotating at 15,000 RPM. It will be supported on Fluorogreen bearings. To keep the dust out, Teflon seals have been made from sleeve bearings with a light press fit, which are mounted either side of the Fluorogreen bearings.

For transporting the soil samples, they are experimenting with Teflon nuts running on an anodized aluminum thread.

A pick may also be used to sample the soil. It is an accordion-like device having a clam shell scoop at one end, one portion of which is extended with a carbide tip to act as a pick. The other half of the scoop is powered with a stepper motor and gear box to enclose the sample that is scraped up. All of the accordion bearings are called out as Fluorogreen bearings. There will be 16 to 20 of these on the pick extension. The pick will strike the Lunar surface with approximately 1 lb. force.

The spec for the soil sampler is written for a duration of two Lunar days.

3.11 Antenna Solar Panel Positioner:

In addition to that contained in the design review outline (Ref. 3) is the following information:

a. In addition to the Ledex solenoid stepper motor, a Landair and Curtiss Wright stepper motor are being considered. They are also considering a digital stepper motor.

b. Bearings in the motors and gear boxes are a Teflon O-ring type of bearing which operates between two 303 stainless steel races, and is designed to be interchangeable with current, standard miniature ball bearings. They also are considering the use of sleeve bearings made of Fluorogreen and Fluorosint. They believe they can get 86% efficiency in their planetary gear boxes with these bearings.

c. Gears will be lubricated either by Teflon idlers or by Teflonizing the gears to a thickness of one ten thousandth (.0001). These gear boxes are of a planetary type with a reduction of 288 to 1. There are four identical units used on the machine.

d. In one of the gear boxes, one planetary is replaced by a worm drive in order to provide for a locking feature. It is planned to experiment with moly-disulfide, aluminum-stainless steel, OBS coating, and Alcomized stainless steel surface treatment for this worm gear.

e. They are designing for operation during three Lunar days.

f. One motor will operate during the Lunar night at -250°F.

3.12 Jet Vane Actuator:

The jet vane actuator is a servo mechanism for controlling the direction of power at mid-course maneuver on the Ranger vehicle. The actuator consists of a torquer coupled with a Markite potentiometer produced by Markite Products Corporation, 155 Waverly Place, New York, New York.

This potentiometer uses a conductive plastic ring for the resistor. This plastic ring is molded on an insulating plastic disc which apparently is a shelf item. Leads are taken off the conductive plastic ring as required by the particular application.

The wiper shaft is mounted in two bearings that look to be about an R4 and an FB 4. The wiper consists of a nonconducting plastic disc on which is riveted a cantilevered beryllium copper spring whose contacts are coated with precious metal. This is probably rhodium. This unit uses an O-ring for the shaft seal. The unit is pressurized and has been tested in a vacuum chamber and successfully held its pressure for sixty days. It would be desirable to eliminate this O-ring in order to eliminate friction which causes difficulties in the servo loop. Tests are going to be conducted in a vacuum without the seal.

Questions raised by this device are: What happens to the plastic conductive ring in a vacuum? What happens to the plastic insulator in the bayonet type connector which brings the leads out of the potentiometer? What is this conductive plastic? This potentiometer is used because this plastic has shown excellent wear resistance. (Subsequent discussion with Markite personnel revealed that this material is a proprietary item.)

3.13 Photocon Commutator:

The early Ranger vehicle used a commutator manufactured by Photocon Research Products, Pasadena, California. The commutator is a counter similar in construction to a Veeder-Root counter.

The wiper is made of stainless steel (Palonay No. 7). The circuitry in the off-the-shelf item is rhodium over copper. However, this combination hung up in vacuum tests and caused miscount. This was corrected by putting 15 millionths gold plate over the rhodium. These tests were conducted at 5×10^{-6} mm Hg vacuum.

The gears which index tens, hundreds, thousands, etc. were brass rotating on stainless steel shafts in the original unit. In vacuum tests the gears seized on the shaft. The first correction tried was to Teflonize the shaft with .0004 thickness of Teflon, using stainless steel gears. The Teflon peeled off the shafts in sheets after 85,000 turns (9,000 revolutions) at $5\frac{1}{2}$ pulses per second. A change was made to stainless steel shafts and Nylatron gears. This combination successfully completed 13 million counts without any indication of difficulty.

The ratchet and pawl arrangement which initiates each count is also made of Nylatron.

4. RELIABILITY

The trend in these programs is toward increased complexity. In the case of the Surveyor and Prospector, advantage will be taken of the technology developed in previous Ranger vehicles in order to minimize developmental risk in components and subsystems. However, each of these vehicles represents a step-by-step advance beyond the preceding vehicle and in each case becomes more complex in order to handle a larger variety of more complicated missions. Thus, even though the basic problem of bearings, gears, hinges, and other sliding surfaces may be essentially the same, increased reliability becomes exceedingly important because of the dependence of a successful mission on the satisfactory performance of an increased number of these parts.

There is an unique aspect of the problem of reliability which is emphasized over and over again. During the next ten years there will be just five opportunities to launch a space vehicle to Mars. Similarly, one can shoot for Venus only once every 19 months. This means that development schedules must be adhered to. It also means that wherever possible, designs must be based on contemporary advanced developments rather than on future development. This factor must be given very careful consideration in whatever research and development program finally evolves for solution of frictional problems.

Also of interest is the fact that when a launch period is available, a launching opportunity to either Mars or Venus consists of roughly a one to two hour period each day for approximately one month. Thus, in the case of Mars, a slip in schedule of one month means a delay of two years. Launching opportunities to Mars will occur in 1962, 1964, 1966 and 1968. Launching opportunities to Venus will occur in 1962, 1964, 1965, 1967, 1969 and 1970 (Ref. 2).

Coupled with this aspect of reliability is increasingly long life requirement. A trip to Venus takes three to four months; to Mars, seven to eight months; and to Jupiter, two years.

Limitations of time and cost add to the need for reliability. While several flights are scheduled for each vehicle series, each flight contains unique experiments which probably cannot be repeated. For example, the friction experiment is scheduled for Ranger 1 and 2. Failure of these two missions may end the opportunit to obtain frictional data during the Ranger series (Refs. 6, 7, 8).

A distinction is made between primary and secondary units in spacecraft. Primary units are those which are vital to accomplishing the over-all mission such as main power sources, telemetry, and guidance. Secondary units are usually individual experiments which, if one fails, do not destroy the value of the mission.

Primary units are usually associated with the basic structure of the spacecraft. Consequently, there is usually more freedom insofar as weight and space are concerned, than is granted to secondary units. Secondary units are usually assigned a block of weight and space and must accomplish their mission within those limits. Consequently, in secondary units, lubrication problems may have to be compromised in order to meet other requirements.

5. LIFE AND STORAGE REQUIREMENTS

Life requirements generally fall into three categories:

5.1 Short Life-Short Environmental Storage:

Such things are booms, solar panels, antenna hinges, and covers fall into this category. Many of these devices will unfold in the parking orbit and motion ceases when flight position has been attained. In some cases high RPM may be involved in this category in switching devices.

5.2 Long Environmental Storage-Short Life:

Secondary experiments usually fall into this category. The unit is required to remain inactive for a long period of time during space flight after which it is required to operate for a relatively brief period while the experiment is being performed. With this type of unit there is a minimum of ground check-out.

5.3 Long Life Continuous Operation:

This is the long duration, long check-out, continuously operating type of unit involving no storage except for ground shelf life. The primary units such as telemetry and guidance would fall into this category.

One cannot limit any of these categories insofar as load and speed are concerned. Also, in general, torque is very limited because of limitations in power supply.

Ground shelf life may be relatively long. For example, parts for the entire Ranger series may be ordered and delivered at one time.

6. ENVIRONMENTAL CONDITIONS

6.1 Temperature:

Spacecraft are usually in the sun and temperature is well controlled. Electronic equipment, such as contained in the Ranger bus, is held near 20°C. Temperature control is maintained on the Ranger vehicles without using moving parts or fluid, by the selection of proper surface finishes, the use of coatings on boxes, and by controlling the spacecraft internal heat transfer capability. On Mariner A vehicles it was planned to use shutters to control the amount of sunlight which the units receive.

At the extremeties of the spacecraft, temperature variation increases. Following are examples of the temperature range expected on various Ranger components:

Electronic assemblies in the bus 0 to 60°C

Solar panels -50 to 100°C

Attitude control jets and regulator 0 to 50°C

Lyman-Alpha telescope detector 10 to 50°C

Friction experiment 10 to 50°C

In the Surveyor spacecraft, temperatures on the cold side of the moon may reach as low as -200°F during that period of each revolution of the moon while the unit is away from the sun. This is approximately a 14 earth day period.

High temperatures will be achieved in spacecraft in such units as the SNAP nuclear power system and Ion engines. Temperatures may reach 2000°F and higher. These systems will not be in use until 1965 or 1966, so high temperatures are farther in the future than low temperatures.

It would be desirable to impose a -65°F requirement on units due to air transportation to the launch site. However, it is doubtful if any design will ever be compromised to meet this requirement.

It appears that the extremes of static environment other than landings on planetary bodies will be -65° to +257°F, which is the sterilization temperature.

6.2 Radiation:

Very little background information on the types and levels of radiation that will be encountered in space and planetary environment is available. This is a new subject about which very little is known and in fact, the initial space vehicle experiments are directed almost entirely at answering these questions.

Ranger 1 and 2 spacecraft will carry a family of radiation detectors designed to monitor the intensity of charged particle radiation (protons and electrons) from energies of about 10^1 power to about 10^8 power electron volts. Ranges of instruments are chosen so that studies can be made of quiet conditions in interplanetary space as well as the phenomena associated with solar disturbances. Ranger 1 and 2 experiments are reported in JPL TR 32-55 (Ref. 4). In addition to the radiation detectors Ranger 1 and 2 will carry a magnetometer to determine the eir planetary magnetic field and its relation to particle flux. Other experiments include a telescope sensitive to Lyman Alpha radiation, a cosmic dust detector, and scintillation counters, to investigate the statistics of solar X-ray.

Mariner 1 and 2 will investigate cosmic radiation, solar corpuscular radiation, and trapped particles near Venus. These launchings are scheduled for 1962. A description of the detectors for use on Mariner vehicles is contained in SPS 37-8 (Ref. 6).

The following conclusions may be drawn at this time:

a. Ultraviolet radiation will definitely be encountered in relatively large quantities.

b. Consideration must be given to micro meteorite bombardment.

c. The effect of secondary radiation resulting from the impact of primary radiation on materials should be investigated.

d. Care must be exercised in drawing any conclusion from experience with nuclear reactors.

(A report on the effect of radiation and vacuum on material properties has recently been issued from JPL.) (Ref. 9)

6.3 Magnetism:

On vehicles incorporating magnetometers, which includes vehicles for some missions of all classes, it is desirable to use nonmagnetic material. In the Ranger a compromise has been made in the use of 440C as bearing material. However, on future vehicles it would be desirable to use more nonmagnetic material.

The permissible magnetic field for the Goddard developed magnetometer, which is in use on several of these spacecraft, is one gamma at one foot.

6.4 Vibration:

Vibration tests are enumerated in Ref. 10.

On type approval tests of Mariner gear boxes, no bearing failures have been experienced due to vibration. Belleville washers are used to preload the bearings. Sligh marks were visible on the bearing race under 20X magnification. However, these marks gave no indication on noise test. It is not certain that the preload on the noise test was the same as the preload induced by the Belleville washers.

There was no sign of fretting corrosion on these bearings.

On the Tiros satellite, (Ref. 66) all bearings are rotated during take-off because of experience with fretting corrosion. On the Mariner R there is insufficient powe available to permit this approach.

6.5 Sterilization:

Sterilization is required of all flight components. For internal sterilization, spacecraft components are subjected to a temperature of 257°F for 24 hours. In the assembly process, surfaces of components are swabbed with a liquid sterilant just prior to mating. Spacecraft external surfaces are exposed to an ethylene oxide gas mixture for a 24 hour period prior to launch. This gas mixture is composed of 12% ethylene oxide gas and 88% freon gas of 25% humidity.

If sub components are not capable of withstanding the high temperature sterilization treatment, it may be necessary to sterilize by means such as radiation. Emphasis in sub components, therefore, should be placed on the development and use of parts which can withstand the required high temperatures and can, therefore, be sterilized by the simpler heat treatment.

6.6 Lunar Environment:

JPL TR 34-159 (Ref. 11) describes the Lunar environment. This is the report around which Prospector proposers are basing their design.

This report cites two summaries of presently accepted knowledge of the moon. These are:

a. "Preliminary Revised Chapters for the Planet", by Harold C. Urey (Yale University Press, October 15, 1959).

b. "The Known Physical Characteristics of the Moon and the Planets", by C. C. Kiess and K. Lassovzsky, (Air Research and Development Command Technical Report No. 58-41, ASTIA Document No. AD 115-617, July 1958).

For purposes of design study, the JPL report assumes the following conditions to exist which are pertinent to the friction problems:

a. The effects of certain factors in the environment will be small and shall not be considered as design conditions. These are:

   1. meteorite particles
   2. magnetic and electric fields
   3. electromagnetic radiation, except for the range of values that effect heat balance
   4. corpuscular radiation

b. The Lunar atmosphere is a vacuum.

c. The Lunar surface temperature falls from a maximum of around 400° Kelvin at the subsolar points to a night time value of approximately 120° Kelvin. The temperature deep in the Lunar interior is 234° Kelvin. The thermal conductivity of the Lunar material is less than 10^{-4} calories per centimeter second per degree Centigrade.

d. Surface dust conditions are uncertain. Although it is necessary to consider this effect, it should not be considered a primary design condition. The surface layer of the moon consists of loose particles approximately three tenths millimeters in diameter composed of material with a high silica content which is a very good insulator.

6.7 Space Vacuum:

Beyond 4000 miles, the vacuum in space is less than 10^{-12} TORR.

7. SIMULATION OF ENVIRONMENTAL CONDITIONS

7.1 Environmental Facilities at JPL:

In the latter part of 1960 a construction program was begun on a simulated space environment facility at JPL. There is a contract with Consolidated Vacuum Corporation, Rochester, New York, to design and build a large space simulator for testing complete

Lunar and planetary spacecraft. This simulator will provide an unobstructed spherical test volume 25 ft. in diameter and will contain an access door 15 x 25 ft. The vacuum pumping system will be capable of operating the chamber at 10^{-6} mm Hg or lower, with a test article gas load of up to 0.165 milliliters per second.

The radiative heat sink consists of a liquid nitrogen cooled liner which blankets the interior wall surfaces seen by the unit under test. Internal optical elements are also cooled to provide radiative characteristics equivalent to those of the shroud.

Radiation approximating the solar spectrum in the 0.2 to 3 micron region and at intensity levels ranging from that at the orbit of Mars to that at the orbit of Venus is to be provided initially in a 15 ft. diameter cylindrical volume concentric with the vertical axis of the chamber. Approximately one-half of the radiant energy will be columnated to $\pm 2\frac{1}{4}°$ with extreme rays within $\pm 4\frac{1}{2}°$. Uniformity is to be ±10% from a mean value in the central horizontal plane and ±15% in the extremes of the illuminated part of the test volume.

The solar simulator portion of the facility has been subcontracted by CVC to Bausch and Lomb Optical Company and is being developed around $2\frac{1}{2}$ kilowatt mercury xenon compact arc lights.

A total of eight 18" x 30" vacuum bell jars are in use or on order. These systems were specially built by CVC and use a TMC 1440 diffusion pump with liquid nitrogen cooled chevron baffles over the diffusion pump. This system should be capable of holding a pressure of 5×10^{-7} mm Hg vacuum over periods of several weeks. It is considered that these systems will not be sufficiently clean and, consequently, modifications may be required.

collimator

An unique concept of producing solar radiation within a vacuum chamber has been experimentally demonstrated. Light from xenon lamps was collected and columnated. Experiments were conducted using a 4 ft. diameter section of a surplus search light parabola which resulted in a columnated (6 to 7°) light beam with a ±2 to 3% variation in intensity over a broad annular test area. Further work is planned to achieve a better definition of the spectro-distribution and to obtain a more uniform radiation intensity across the entire circle (Refs. 6, 7, 8).

7.2 Environmental Equipment at Hughes Aircraft:

For vacuums to 10^{-7} mm Hg, Hughes is using both mercury and oil diffusion pumps. The mercury pumps are some that they have made themselves. For vacuums to 10^{-9} mm Hg, they are using a Varian Associates Vac Ion Pump. They do not use cryogenic pumping with this vacuum system.

They have one chamber which is capable of going to 10^{-12} mm Hg in about 100 cubic centimeters of space. This system utilizes two mercury diffusion pumps in series and two cryogenic pumps in series.

In general, they use glass enclosures for their vacuum work. They have three expert glass blowers who construct enclosures specifically for a particular test. As a consequence, the glass enclosure is essentially molded to the shape of the test assembly resulting in a minimum of unused volume within the enclosure. One advantage gained from the use of glass is that it permits visual observation of vapor deposition when testing starts. Sometimes vapor deposits disappear as the rig is dismantled and would give no visual indication of having been formed.

Hughes has a patent pending on a high torque drive unit. This consists of a differentially pumped mercury trap incorporated on the drive shaft. The drive is mounted vertically with a cup incorporated integral with the drive shaft. This cup rides on the O.D. of a ball bearing. The I.D. of the bearing is supported on the vacuum enclosure. The cup is filled with mercury and a pressure is maintained on the outside of the cup approximately equal to the vacuum inside the chamber. Above the level of the cup is a nitrogen cold trap in the vacuum chamber housing to trap mercury vapor.

For solar simulation they use a water cooled GE bulb DH6. This is an off-the-shelf item.

They expressed concern over being able to duplicate experimentally the effects of thermal shock which are expected to occur between the Lunar day and the Lunar night. While, in general, the operation of Surveyor may occur only during the Lunar day, it is expected that some operation is expected to continue into the Lunar night before shutting down. The extent to which this is done will depend on power supply.

7.3 Problems of Simulating Space Vacuum:

There is concern at JPL, as elsewhere, about the validity of vacuum in laboratory experiments. Mr. Rittenhouse is familiar with the NASA report (Ref. 58) and also referred to a report from Goddard Space Flight Center, "Preliminary Test Results of Bearings for Vacuum Operation", by Harold E. Evans and Thomas W. Flapley. This report discusses their test equipment which uses molecular sieve to achieve purity of vacuum. Mr. Rittenhouse is, at the present time, investigating equipment requirements.

7.4 Flight Experiments:

Due to the problems of environmental simulation, it would be desirable to obtain test data from actual space mission. A separate friction experiment is scheduled for Ranger 1 and 2 and Dr. F. J. Clauss at Lockheed is working on a bearing test rig to be orbited in a satellite. One idea is that many of the experiments aboard space vehicles which are of a relatively short duration could provide useful friction information if they were allowed to continue to operate beyond the life of the experiment and monitor back information on bearings, gears, and the like. This information would be useful even if it were only to indicate that units continued to operate.

It was indicated that there may be a problem from the standpoint of power that would prevent continued operation of experiments. It was also indicated that there may be limitation on the ability to transmit information to indicate bearing performance.

Everyone contacted is approaching the problem of space simulation by trying to duplicate the conditions which exist in space. Apparently little thought has been given to the use of controlled atmospheres (inert gases) to provide conditions under which results would correlate with vacuum.

8. PRESENT JPL LUBRICATION PRACTICE

8.1 Bearings:

Grade 7 bearings are used with phenolic retainers impregnated with General Electric Versilube F50 oil. Lubrication is supplemented by a thin film of General Electric Versilube G300 grease on the races. Bearings are installed in sealed chambers. The seal consists of a double O-ring of Viton A (Parker Compound 77-545) material with the space between filled with GE 300 grease. Bearings are equipped with shields whenever it is possible to procure such (Ref. 12, 13).

Bearing inspection is subcontracted to Bearing Inspection Corporation. Here they are ultrasonically cleaned, run-in in a soap and water solution, and cleaned again. They are then tested on a noise analyzer developed by Bearing Inspection. Bearings are rejected unless they show a completely random noise pattern. A high rejection rate (50%) has existed.

Bearing Inspection also provides a service in filtering the GE 300 grease to 10 microns. They lubricate the bearings with grease and supply syringes for additional lubrication in spaces adjacent to the bearing.

8.2 Gears:

Gears are in general lubricated with a bonded solid film lubricant. Phenolic bonded molybdenum disulfide (Electrofilm 4396) is usually used. Worm gears and other gear types having a large amount of sliding contact are avoided due to the excessive wear and the inability of the dry film coating to withstand the sliding contact.

9. MISCELLANEOUS COMMENTS

A need is expressed for a "Marx Handbook of Space". This would include:

a. A list of weldability of materials similar to the galvanic series.

b. The effect of long storage in vacuum environment on the adherence of materials to each other, such as O-rings welding to metallic surface.

c. A list of wear rates of various commonly used materials rubbing on each other without lubrication.

d. A list of coefficients of friction of various materials rubbing on each other with no lubrication.

e. The effect of running speed, temperature, and bearing pressures on friction coefficients of materials in space environment.

f. The effect of space environment on material physical properties.

The feeling is expressed that designers are required to make too many calculated risks. It has been suggested that more extensive mechanical experiments be conducted in space (like the friction experiment on Ranger 1 and 2) in contrast to scientific experiments.

From the designers' standpoint, the primary problems insofar as bearings are concerned are:

a. Torque consistency and low torque

b. High speed

c. Weight reduction - smaller gears and bearings, smaller motors (less torque requirement), elimination of hermetically sealed cases.

The technique of using barriers to stop creep will be difficult in miniature electrical components. For example, a size 8 motor is 3/4 in. in diameter and 1/8 in. long and the interior largely taken up with wire.

Hermetic seals are not available at high torque level (100 in. lbs.).

In general, the delicacy of most instrument experiments require that some enclosure be provided even if it is not hermetically sealed.

Cases are in general, heat sinks. Heat conductance from motor armatures must be through the bearing. Consequently, at cold outer temperatures the temperature difference across the bearing is increased. This must be given consideration in establishing bearing internal clearances.

Viton A material is called out for all O-ring seal applications, for the following reasons: permeability, sterilization temperature requirements, and radiation resistar

A workable worm gear would be very desirable because of the large gear reduction usually required with AC motors.

To date there has been no use of ball screws aboard spacecraft. However, it is felt that there will be application for ball screws in the future. These would be of the order of a 1/4 in. diameter screw with as fine a pitch as possible (very light loads).

It was suggested that stearates be investigated for dry film lubricants.

It was suggested that wooden bearings, impregnated with oil be considered for some applications (lignum vitae).

10. ELECTRIC MOTORS

The principle source of torque on board the spacecraft is likely to be a small electric motor which can operate directly in the high vacuum of space and is self-lubricating. Power is generally in the form of 400 cycles current using synchronous motors operating at speeds of 6000 RPM or greater. This has two disadvantages: there is little need for 400 cycle current elsewhere (the solar panels generate DC directly) and the desired output speed usually requires a complicated gear train. There is thus a need for low speed DC motors and undoubtedly considerable development work will be done in this area.

Motors will often find application involving continuous operation for extremely long periods (1000 hours on up) and, consequently, grease and oil must be candidate lubricants. This gives rise to a possible need for sealed bearing development versus the shielded bearings now in use. However, torque is critical in these applications. An area for further study is the use of external grease chambers designed into the bearing housing.

Lubrication of antifriction bearings in motors now uses phenolic retainers impregnated with GE F50 silicone oil. Consideration should be given to improved retainer materials capable of increased impregnation. Consideration should also be given to increased oil capacity through the use of integral shaft lubrication and porous ring impregnation.

A typical example of an AC, synchronous, hysteresis type motor is the following:

Power input	-	2 watts
Motor speed	-	12,000 RPM
Integral gear reduction	-	1500 to 1
Number of gear shafts	-	6
Bearing size	-	.070 bore (approx.)
Type of gearing	-	spur gear, 120 pitch, 20° pressure angle

One source for AC motors is Gaylord Reeves Company, Pasadena, California. They have known their motors to operate at 5×10^{-6} mm Hg for up to 1000 hours in vacuum chambers. Their motors are supplied primarily with Barden, Microtech, and Reed bearings. Usually the lubricant is specified by the customer. They have furnished bearings lubricated with silicones, molybdenum disulfide and CBS coating.

They feel that AC motors will always have to fill the need for synchronous speed control since there is no way at present to get around the speed regulation problem in DC motors. Otherwise, the choice between AC and DC is largely a choice between gears and brushes. To their way of thinking, the ratchet type of DC motor merely adds another friction problem to the bearings already present. However, they do concede that there may be future space application for magnetically detented DC motors in which the field is rotated with solid state devices.

Bearing torque is extremely critical in a synchronous AC motor. This type, being essentially a constant torque device, gives peak efficiency with increased speed up to the stall point. Development effort is always directed at increasing the stall level to get maximum power from a given package. Thus, the level of bearing torque at the design point is especially critical and in some instances, because of the relatively low starting torque, bearing starting torque may be extremely critical. This is in contrast to the DC type where a high starting torque is available, dropping off with speed and resulting in peak efficiency in the mid-speed range.

DC motors are a more recent development than the AC type and, consequently, have not come into extensive use. There is, however, a strong tendency to use this type since speeds are relatively low. DC motors fall into two classes: one has a ratcheting mechanism and the other achieves DC brushless performance with a switching field.

At low frequency these motors move in discrete steps and this would have to be taken into account in selecting bearings. At high frequency the motion can be made to approximate continuous rotation. At these high frequencies, however, it is almost essential to have over shoot in order to lock in at the next step. This over shoot and skidding results from a back torque to lock in at the step. They feel that bearing life is cut in half by this condition over what it would be for continuous running.

One DC motor manufacturer is IMC Magnetic Corporation, Maywood, California. They make stepper motors with logic controlled fields. Their general production line of motors have 90° steps per pulse at 200 pulses per second. They have made special motors going as high as 500 pulses per second. They are working on units that would have three degrees per pulse. These units will have correspondingly high torque and it is felt will meet a real need for slow speed high torque power, eliminating gear boxes (Refs. 30, 31).

11. ADVANCED LUNAR VEHICLES

11.1 The Prospector vehicle will introduce the problem of dust and grit on the Lunar surface.

It will also introduce increases in the size of components, larger loads, transportation, hoists, and other equipment handling requirements including increases in duration of component operation. A vehicle is desired which can operate for a year or more. The minimum operating time will be one Lunar day and night.

Power will be extremely critical, especially during the Lunar night, so that the possibility of heating lubricants to survive the extreme cold of the night is extremely doubtful.

Apparently the Prospector program will quickly become involved with manned Lunar operation. This introduces a question with respect to existing or operating during the Lunar night. Those who are primarily concerned with equipment design take the approach that equipment will operate during the day and exist during the night. Power for daytime operation will be derived from solar cells. Man, however, is expected to work more easily during the Lunar night and the problem is to permit him to survive during the Lunar day. This is due to the effect which daytime solar storms will have on human tissue during Lunar day exposure. Apparently man can be warmed during the night more easily than he can be protected from proton bombardment during the day.

The Prospector vehicle will use electrical power distribution. There are no hydraulic nor pneumatic systems contemplated. The source power will be largely from solar panels. However, there is a possibility that nuclear power systems or fuel cells will be developed in time for this program.

Exposure of components on the Prospector vehicle to vacuum environment will be almost a necessity in contrast to the general evasion of this problem on the Ranger and Mariner vehicles.

The Prospector vehicle will be contracted as is the Surveyor. While only one Prospector vehicle design is anticipated, this design may be combined with that of the Apollo vehicle so that the same bus can serve both programs. There is a possibility that two sizes of Prospector may be designed for scientific missions, coinciding with the C2 and C4 boosters.

Apollo is now set for 1967. To coincide with this schedule, the Prospector must know how to do its job by late 1963 and be performing missions by late 1964.

11.2 Some Lunar exploratory vehicles are being studied which may find application to the Surveyor program. One such prototype vehicle has been constructed and tested by Space General Division, Aerojet General Corporation, Glendale, California (Ref. 1).

Tests to date have been directed primarily at development of the electronic equipment and the basic mechanical design while operating in earth environment. Only a limited amount of testing has been done in vacuum in an attempt to develop a shaft seal for the main motor power to the leg.

Their main problem is to get power to the legs from a rotating motor. They wish to seal the motor shaft to permit pressurizing the interior of the motor case to 7 or 8 lbs. per square inch opposing a vacuum of 10^{-12} mm Hg at the Lunar surface. They have succeeded in testing a seal at 7 psi in a vacuum of 10^{-7} mm Hg for approximately 8 hours. This is a wedge type Teflon seal. Attempts to use Rulon and Flucroscint resulted in leakage, which, though small, was great compared to the Teflon. However, the Teflon seal also began to leak appreciably when temperatures reached 120°F.

Design of the Lunar walking vehicle is following the JPL standard moon criteria contained in JPL TR 34-159 (Ref. 11). They are seeking 200 hours operational life which is made up of 8 hours per earth day over a period of two Lunar days. The vehicle can only operate while it is in line with the Goldstone tracking station to permit communication between the vehicle and earth.

Leg movement is accomplished by cams and rocker arms. In the prototype, cables were used from the rocker arms to the leg but is being discontinued due to excessive cable stretch. In future models, cables will be replaced with rods. The load on the cam is about 200 lbs. Needle bearings are presently being used for the cam follower.

The antenna and solar panel mechanism is powered by electric motors and gear boxes much as described in previous reports of Ranger and Surveyor vehicles.

The front pair of legs is hinged to permit movement from side to side to steer the vehicle. The knees on the legs are oscillated in miniature ball bearings of approximately 3/16 in. bore. These bearings are mounted in a yoke with a pin supported in their bore. Bellows are being considered to seal the leg knees and other joints.

The reasons cited for using legs rather than wheels are:

a. Require less power to walk than to roll in the sandy surface of the moon.

b. Leg extension gives a broad, stable wheelbase and provides for easier maneuvering over obstacles.

c. Wheels would have to be of a radius approximately equal to the leg length and would be extremely heavy and difficult to fold into a suitable launching position.

12. PLASTIC BEARINGS

Plastic bearings are scheduled for use on the Mariner R vehicle in the form of Fabroid (Teflon-glass) monoball type bearings (Ref. 51) and DU spherical bearings (Ref. 50).

No difficulty has been experienced with Fluoroscint bearings except for one brittle fracture which is considered a fluke. Fluoroscint bearings were called out for use in louvers in the Mariner A vehicle. Although the Mariner A has since been cancelled, these bearings were run successfully through the type approval test.

The question of heat transfer using Teflon sleeve bearings has been raised. In tests which have been conducted at 10^{-5} mm Hg and 6000 RPM on size 11 electric motors, bearing temperatures reach 300°F in about 19 minutes. This temperature was measured by a thermocouple in a hole in the armature shaft. At 70 RPM with DU bearings, temperature reached 325°F in a relatively short time and terminated the test. As a result of this, a program of extensive temperature measurement has been initiated. A belief has been expressed by several people at JPL that DU metal will solve the heat transfer problem due to the metallic backing which is used to support the Teflon.

Barden has indicated that a maximum of 2 lbs. load can be carried on R2 to R4 size bearings with Bartemp retainers. These bearings operate at very low speeds of the order of 150 RPM or less.

12.1 Fabroid Bearings:

A discussion was held with representatives of Micro Precision Division, Micromatic Hone Corporation, Los Angeles, California, manufacturers of Fabroid bearings (Ref. 51).

Fabroid bearings consist of a Teflon-glass fiber weave backed by a phenolic impregnated, glass woven backing bonded under heat and pressure to a phenolic journal. The two layers of glass are bonded to each other and to the base material simultaneously. Subsequently, the assembly is heat treated on a mandrel to press to size.

These are available as both rod end and journal type bearings. Rod end bearings can be loaded to 60,000 psi and on journal bearings, a maximum design value of 20,000 PV can be tolerated. Tests are now being conducted up to 30,000 PV. A maximum surface speed of 150 feet per minute is recommended.

Fabroid has been bonded to 440C stainless steel and aluminum, as well as phenolic.

Fabroid bearings are scheduled for use as rod end bearings supporting the radiomete on the Mariner R vehicle. In this application alignment is not particularly critic

The minimum sizes available are 3/16 in. sleeve and 1/8 in. rod end. The 3/16 in. sleeve is available only in molded phenolic backing.

This company has had no vacuum experience whatsoever.

A coefficient of friction of .04 is quoted for loads of 2000 psi or larger with a 440C shaft at 56 Rockwell C and 4 to 8 rms microfinish. The friction coefficient is somewhat higher at lower loads.

The bearings come in two types: Type 1 has a double layer approximately .0206 whil Type 2 has a double layer (Teflon-glass fiber weave and impregnated glass backing) of approximately .0106.

Although these are basically high load, low speed bearings, they have run at 500 feet per minute with very light load on a servo motor having intermittent service, with very good results. It was also stated that these bearings have been used in cryogenic application with good results.

13. ELECTRICAL CONTACTS

Temperature and vacuum are considered the main problems with electrical contacts. date all units have been hermetically sealed because you cannot buy a good switch wi vacuum capability. It is anticipated, however, that hermetic sealing cannot be con sidered in future missions (for sealed systems a power loss of $500 to $600 per wat is cited and power is expected to be a limiting factor until such sources as nuclea power are fully developed). Thus, vacuum operating capabilities are required.

Vibration, as experienced at launch, is not expected to be a problem due to the very light weight yet extreme stiffness of potentiometer and encoder parts. The spring load of the wipers should be sufficient to maintain contact and position during the short extreme vibration environment.

A sample problem is a linear potentiometer in which the voltage at the wiper should be directly proportional to the position of the wiper relative to the over-all length of the coil and the voltage across the coil. The output of these units, instead of being a straight line, is a line with severe peaks. This condition occurs while operating in sea level atmospheric conditions, While its cause is not known, it is due to an increase in resistance, which may be caused by dirt. The existence of this condition makes one apprehensive toward vacuum performance.

I could locate no one who could give me electrical requirements that would be imposed on candidate materials for electrical contacts in vacuum. In general it was difficult to get any information on the subject of electrical contacts, but it appears that this is a subject worthy of extensive investigations. Micro switches which are used have gold contact. It is believed that these gold contacts actually weld when contact is made and are broken apart by spring load when contact is terminated .

14. SPACE FRICTION EXPERIMENT

An experiment specifically designed to provide information on frictional behavior in [illegible] space vacuum has been scheduled to fly on Ranger 1 and 2. This experiment and preliminary laboratory experiments are discussed fully in SPS 37-8 and 37-10 (Refs. 6, 8)

Because the first Ranger shot did not achieve elliptical orbit, the maximum vacuum achieved was of the order of 10^{-6} mm Hg. The data which were telemetered back during this flight indicates friction levels not too different from those achieved in laboratory tests. These data will be published shortly.

A malfunction in the subsequent firing of Ranger 2, again prevented the desired orbit.

Section B

BEARINGS

1. DISCUSSION

While it is recognized that a potential problem area exists with regard to bearings to operate in the absence of earth's atmosphere, the problems of validly simulating these environmental conditions are so great as to prevent extended testing to date. In the tests which have been performed there has been a suspicion of regeneration of a lubricating film on the parts due to reabsorption of outgassed particles (Ref. 58). This condition has resulted from necessary use of vacuum facilities of finite volume in which such reflection is possible as compared to the infinitely large escape volume of outer space. The deleterious effect of lubricant vaporization and the tendency of materials to weld in vacuum have, however, been suitably demonstrated (Refs. 55, 56).

At the present time, pending tests which are programmed specifically for space applications, we can only analyze these problems on the basis of basic material characteristics, experience in vacuum tube equipment and other applications in which some other extreme environmental condition is substituted for the lack of atmosphere. Development in other advanced fields of application contribute many candidate materials. Jet engine requirements have pioneered the development of several synthetic lubricants, grease development is directed to temperatures of 800 to 1000°F, and considerable effort has been expended in the development of dry film lubricants for extremes of temperature and special atmospheres. Several candidate lubricants are shown in Table 1.

Similarly, current experimental programs have produced not only significant test results but also considerable experience in the fabrication of bearings from nickel base, cobalt base, ceramics, cermets, carbides, glasses, and other unconventional bearing materials.

TABLE 1

OILS				
Product	Manufacturer	Type	Vapor Pressure mm Hg	Remarks
HT 103	E. F. Houghton	Mineral		Evaporation rate at 110°(and 10^{-7} mm Hg, .7 to 1.(10^{-4} gr/cm^2/hour
F-50	General Elec.	Silicone (chlorophenyl-methyl poly-siloxane)	50 at 450°F	Evaporation rate less th. 1% after 3 hours at 200°(
OS-124	Monsanto	Polyphenyl ether	.7 at 500°F	Evaporation rate 1.5% af 48 hours at 550°F
Apiezon K	Biddle	Mineral	10^{-3} at 572°F	
67024	Dow Corning	Silicone	7.6 at 600°F	Evaporation rate 25% afte 48 hours at 550°F
L-743	Sinclair	Synthetic MIL-L-25336		

GREASES			
Product	Manufacturer	Type	Remarks
G-300	General Elec.	Lithium soap-silicone (F-50)	Evaporation rate 1.0 x 1(gr/cm^2/hour at 110°C anc 10^{-7} mm Hg
L-760	Lehigh Chem.	Ester thickened	Evaporation rate 0.1 x 1(gr/cm^2/hour at 110°C anc 10^{-7} mm Hg
EG 518	M-R-C	Polyphenyl ether	
ETR-B	Shell	Silicone thickened	Evaporation rate 1.0 x 1(gr/cm^2/hour at 110°C anc 10^{-7} mm Hg

Experience with actual space flight mechanisms has been relatively limited compared to the eventual goal of our national space program. In these mechanisms it has been possible to circumvent bearing problems due to short life duration and the ability to establish environmental conditions within the range of past experience. This approach while permitting space flights now, adds little to knowledge needed for the future.

Our approach herein has been to recommend those bearing systems which offer promise for space application, on the basis of existing experience with other extreme applications, and to propose further experimental work to fill in those areas where further development is essential. Load, Speed and Duration of flight have been taken as the principle parameters for initial lubricant and material selection.

Before discussing these recommendations in detail, it is in order to relate the space problem to the existing state of the art.

1.1 Sliding Friction in Ball Bearings:

Antifriction bearings contain three elements of sliding contact:

a. Differential speed in contact area - In a ball bearing under pure radial load, pure rolling is accomplished in only two relatively narrow bands located approximately at the centroids of the contact area, on each side of the center line in the direction of rolling. At the center line as well as at the outer extremities of the contact area, differential slip is induced. These incremental sliding components, fortunately, are low in velocity and invoke only low order lubrication requirements.

b. In a ball bearing under axial load, the contact angle causes the ball to race contact to be inclined with regard to the center line of the shaft. This means that the increments of the contact are on on varying radii from their centers of rotation, thereby invoking a rather severe instantaneous twisting of the contact areas. These shearing forces impose a severe requirement on the lubricant, and, in the case of marginal lubrication, have been demonstrated to increase friction and accelerate wear dramatically.

c. Friction of the cage against the rolling element and guiding lands results from differential speed of the members of the rolling element complement as they enter and leave the radially loaded zone. This type of friction is self-regenerative in that deterioration of the surfaces in sliding contact increases the friction and this in turn progresses the deterioration of the surfaces. It has been demonstrated that cage friction, by increasing the rolling contact tangential effort required to rotate the cage-rolling element assembly, can precipitate failure in the area under rolling contact as well as in the retainer contact. In a full type or cageless bearing configuration, friction between the rolling elements is substituted for the cage friction and at higher relative velocity.

1.2 Lubrication:

Lubrication of bearings in conventional applications is based on the simple theory that the rolling and sliding parts will operate most satisfactorily when separated by a thin film of material of low shear strength. The problems associated with high temperature, cryogenic, vacuum, and any other extreme application begin with the inability of conventional lubricants to sustain such a film under these environmental conditions.

As a general rule, the order of preference in lubricating systems is: fluid, grease and dry film.

Oils and greases offer the advantage of low torque. There is a wealth of background experience compared to other lubricant types. The use of oils and greases generally involves relatively simple techniques, compared to the application of film coatings, with attendant increase in reliability.

There are, however, special considerations which must be given to the use of fluid lubricants in space environment. These include: loss of fluid by evaporation and creepage, frictional polymerization catalyzed by clean metal surfaces, effect of absence of oxygen and other gases on the formation of lubricating films, and radiation stability. These factors are discussed at length by Dr. Clauss (Ref. 55).

Oils and greases will continue to find application in space mechanisms, despite these effects. Many objections to the use of such lubricants in space mechanisms can be overcome by design ingenuity. For example, experimentation at MRC has shown the feasibility of utilizing a porous ring in the inner race to supply small quantities of fluid lubricant from an integral shaft reservoir. Grease lubrication may be extended by appropriate design of the grease cavity. Boots can be designed to hold pressure and deter evaporation.

In some cases, extended fluid lubricant life will require development of seals to retard evaporation and to sustain high pressures in the lubricant cavity.

Future space vehicles will make extensive use of dry film lubricants and self-lubricating materials. Experience with high temperature bearing development has shown that friction, wear, and life are dependent on the combination of lubricants and materials in contact. No general rule exists to permit the selection of an optimum material for a given temperature range. In the theory of dry film lubrication the combination of ring and rolling element materi with the retainer material and dry film coating are of importance in the development of a selfregenerating lubricating film. In any dry film lubricated bearing, either the materials present or the atmosphere must provide the reagent required to produce a beneficial oxide from the combination of the dry film and bearing materials. In this connection the composition of the retainer, ring and ball or roller materials is significant with regard to the presence of constituents capable of forming eutectics in combination with the dry film material.

In vacuum, the lubrication problem is aggravated by the lack of atmosphere since the oxides and sulfides generally attendant in the material surfaces tend to diminish, presenting a pure surface to contact. Under these conditions, there is a tendency for materials to weld together. This problem can be attacked in two ways, by separating the mating surfaces as by a film, or by developing materials which are compatible in contact with each other, and thus will not weld.

In general the following materials appear to possess the characteristics desirable for combining with existing lubricants to satisfy the majority of criteria for outer space applications: M-50, 440-CM, Rene' 41, Stellite 6B, S-Monel, Ni-Resist (see Table 2). These materials are high in nickel, cobalt or molybdenum content.

Under marginal lubrication drastically greater friction and wear result from sliding than from rolling. At the present time, pending development of materials and test data, design ingenuity must be directed toward avoiding sliding contact. However, even in sliding, candidate dry lubricants exist for space application (see Table 3).

1.3 Bearing Design:

There is no known bearing design configuration which will eliminate the sliding friction in the contact areas of a ball bearing under axial load. The most obvious solution to this phase of the problem is to eliminate axial loading wherever possible

TABLE 2

MATERIALS								
Designation	Content							
B-100	1.25 C	16 Cr	8 W	8 Co	4 Mo	Bal Fe		
WB 65	1.0 C	16 Cr	4 W	5 Co	1 V	Bal Fe		
Stellite 6B	1.1 C	30 Cr	4.5 W	3.0 Ni	3.0 Fe	Bal Co		
Rene' 41	.09 C	19 Cr	1.5 Al	3.1 Ti	11 Co	10 Mo	Bal Ni	
440-CM	1.15 C	14 Cr	4.0 Mo	0.50Mn	0.50Si	0.25Ni		
440-C	1.15 C	17 Cr	0.45Mo	0.50Mn	0.50Si	0.25Ni		
Stellite 25	0.1 C	20 Cr	1.0 Si	3.0 Fe	15.0 W	10 Ni	Bal Co	
M-50	0.8 C	4.1 Cr	1.1 V	4.25Mo	0.30Mn	0.25Si	Bal Fe	
Ni-Resist	2.6 C	4.5 Cr	5.0 Si	0.4 Mn	29.0Ni	Bal Fe		
S-Monel	63.0Ni	30.0Cu	4.0 Si	2.0 Fe				
Pyroceram	Glass having nucleating agents							
Tungsten Carbide CA-3	94.0WC	6.0 Co						
M-50 M (Carburized)	0.8 C	0.31Mn	0.25Si	4.0 Cr	1.0 V	4.25Mo	Bal Fe	
H-12 (Nitrided)	0.35 C	0.29Mn	0.77Si	5.0 Cr	0.24 V	1.27 W	1.32Mo	Bal Fe
Rulon	Glass impregnated Teflon							
Armalon	Fluorocarbon resin coated glass fabric							
Fabroid	Glass-phenolic reinforced Teflon							
DU Material	Metal reinforced Teflon-lead mixture							

TABLE 3

DRY FILMS	
Type	Temperature Range
PTFE - Teflon	-430°F to 350°F
MoS_2 - Phenolic Epoxy Bond	-300°F to 700°F
MoS_2 - Silicone Resin Bonded	-300°F to 600°F
MoS_2 - Sodium Silicate Bonded	-365°F to 1200°F
Gold Film	-65°F to 900°F
Metal Free Phthalocyanine	room to 950°F
CBS CDL-5940	room to 700°F
Electrofilm #1000 (Graphite Ceramic Bond)	800°F to 1500°F

by balance of design forces. In those cases where this is not possible, palliative measures may be taken in the design of the bearing. Adjustments in internal geometry to minimize the contact stress and the length of contact will reduce friction resulting from twisting of the contact areas. Minimizing contact angles also tends to reduce the twisting of the load carrying contact area. It may be advisable, in some instances, to separate radial and thrust load, to be carried by separate bearings.

Experiments with radially loaded bearings indicate that sliding friction can be practically eliminated by the incorporation of quills between the rolling elements so designed that the quills are carried as a separate complement on their own races. In this design relative velocity between the quills and the load carrying rolling element is reduced to zero (nominal).

2. BEARING APPLICATIONS (See Table 4)

Our analysis of bearing applications in the field of outer space mechanisms indicates that four general categories may be considered. While some bearing applications may overlap more than one of these categories, the following recommendations may prove of use.

In the first category we have defined the operating conditions as high load-low speed. In the second category we have considered the low load-high speed applications, while in the third category the low load-low speed condition is included. The fourth category covers oscillating bearings, in which the rotation is less than 360°. In many applications, bearings of this type are quite highly loaded.

In order to roughly define the speed and load areas considered, the following parameters are presented:

	S	C/P
Very High Speed	Above 500×10^{10}	Above 100
High Load-Low Speed	Below 128×10^{10}	Below 10
Low Load-High Speed	128×10^{10} to 276×10^{12}	Above 40
Low Load-Low Speed	Below 128×10^{10}	Above 15

$S = N^3 d^3 D$

N = Speed (RPM)

d = Ball or roller diameter (in.)

D = Bearing pitch diameter (mm)

$$= \frac{\text{Bearing O.D. (mm)} + \text{Bearing Bore (mm)}}{2}$$

$$C/P = \frac{\text{Bearing Capacity @ 33 1/3 RPM (*)}}{\text{Applied Load}}$$

*from manufacturer's catalog, or derived from AFBMA data

2.1 Two Hours Continuous Operation at Start of Mission:

Within these categories we have considered, first, the service requirement of approximately two hours of operation at the beginning of a mission. These mechanisms will be subjected to atmospheric conditions ranging from normal earth's atmospheric pressure to high vacuum.

2.1.1 High Load-Low Speed:

For the high load-low speed category we recommend the use of oil lubrication, applied to ball or cylindrical roller bearings of M-50 high speed tool steel, fitted with silver plated bronze retainers.

This suggestion is based on specific experience gained in liquid fuel pumps for rocket motors such as designed, manufactured, and tested by Rocketdyne Division. This experience included the operation of bearings at high speeds and under high specific loads and included the development of lubricants and lubrication systems to overcome the problems of foaming at high altitudes.

As an alternate, grease lubrication may be used with either ball or cylindrical roller bearings. Again we recommend that the ring and ball or roller material should be M-50 steel. In this case we would suggest that the retainers be manufactured of S-Monel.

While grease lubrication has demonstrated its ability to lubricate bearings fitted with silver plated bronze retainers, we have found that a reduced section retainer design can better be accomplished by the use of S-Monel. In grease lubrication the mechanism of grease motion and transfer to provide lubrication to the critical bearing elements is of great importance. A properly designed retainer for grease lubrication will not only be of minimum section but will be contoured to promote the flow of grease from the external surfaces of the bearings into the ball or roller complement. In general, a grease lubricated bearing will generate more heat than an oil lubricated bearing. Perhaps this could be stated in another way, that with oil lubrication heat will be carried away from the rolling element contacts faster than with grease lubrication. The S-Monel retainer material has a higher limiting temperature than silver plated bronze retainer and, therefore, for this reason also is a better selection for this category.

As a second alternate we recommend the use of ball or cylindrical roller bearings with rings and balls or rollers of Rene' 41 and retainers of Ni-Resist. These bearings should be lubricated with NAMC AML-23A dry film applied to the retainers only.

In any bearing, heat generation and stabilized temperature are functions of load and speed. The use of a dry film lubricant results in a higher friction torque than obtained with oil or grease lubrication. Even with the relatively low ambient temperatures assumed for this study, this bearing, operating under high load, must be assigned to the category of high temperature bearings. Therefore, our experience with high temperature ball and roller bearings operated in air becomes pertinent. The materials indicated above have been found to operate reliably, well in excess of the two hour service requirement, when incorporated into bearings of appropriate design. These design criteria include exceptionally large clearances, both bearing internal clearance and clearance of the retainer with its guiding land and with the balls or rollers. In addition, such bearings must include in their design, consideration of escape paths for wear debris. For ball bearings subjected to thrust loading open conformity must be used, combined with the large clearances mentioned previously, requiring, in turn, exceptionally large race shoulder heights to retain the contact areas within the races (Ref. 111).

Care must be exercised to avoid rigid axial preloading of such bearings. Since a high temperature bearing outer race cannot be depended upon to slide in a housing, it is necessary to exercise design ingenuity to introduce spring preloading through diaphragm mountings or similar devices.

Usually it is more convenient to mount one ball bearing and one roller bearing on the shaft to be supported. If the thrust is unidirectional at all times this arrangement is simple. However, if the thrust reverses during operation, the single row bearing must be designed to support thrust in both directions, or the opposed mounting arrangement discussed above must be incorporated. A bi-directional thrust bearing incorporating the features listed above can be manufactured, although the design of the wear debris escape paths presents considerable difficulty, with this design.

2.1.2 Low Load-High Speed:

For the low load-high speed category, still in the short service life requirement, we recommend the same bearing as our first choice for the preceding category: a ball or cylindrical roller bearing with rings and balls or rollers of M-50 steel. The retaine should be of silver plated bronze, except for very high speeds, in which case retainer should be of bakelite compound material.

This basic recommendation is again based on experience with rocket engine fuel pump drives, proving the practicality of the lubrication system, and is substantiated by thousands of successful applications of high speed bearings, including jet engine main shaft and accessory applications.

In the very high speed range, many product development and bearing research programs have demonstrated the superiority of the bakelite compound retainer. With oil lubrication and within the temperature limitations of this material (approx. 300°F) the use of the bakelite compound retainer permits higher speed operation than possible with any other known retainer material. Its light weight and oil wicking and retention properties combine to give this material these excellent properties.

In this category as an alternate we recommend grease lubrication with the same M-50 steel ring and ball or roller material. In this case the retainer should be of S-Monel.

Again, with the introduction of grease lubrication it is necessary to incorporate reduced section, contoured retainers to promote the flow of lubricant into the bearing areas. The bearing operating temperature can be expected to be higher than with oil lubrication. Both of these factors direct our recommendations to S-Monel for this retainer. In view of its relatively low specific strength, the design of reduced section retainers of bakelite is questionable. While many grease lubricated bearings have been fitted with bakelite compound retainers, the bulk of the retainers makes it difficult to promote grease flow, required for reliable bearing lubrication.

However, there may well be specific bearing designs outside the generalized scope of this presentation, which could justify the introduction of the bakelite compound retainer in a grease lubricated bearing.

As a second alternate the ring and balls or rollers should be manufactured of Rene' 41 material and the retainers of Ni-Resist. The retainers should be dry film coated with NAMC AML-23A.

While this alternate must be included to cover those applications which cannot tolerat oil or grease lubrication, it must be recognized that the limiting speed of dry film lubricated bearings is considerably lower than attainable with oil or grease lubrication. That dry film lubricated bearings are applicable for this requirement has been amply demonstrated by programs at Stratos and NASA, Lewis Research Center (Ref. 57).

2.1.3 Low Load-Low Speed:

For the low load-low speed category we recommend the use of grease lubrication in ball bearings manufactured with rings and balls of M-50 steel and retainers of S-Monel

The requirements of this category are easy to meet. In fact, the rings and balls of these bearings could well be of SAE 52100 steel or 440-C stainless steel. However, we feel that a slight improvement in reliability is achieved by the use of M-50 steel and in addition, bearings of this material had a wider range of potential application, beyond the simple requirements of this category, as defined. Again, retainers of S-Monel are recommended to provide a high strength material, permitting design latitud necessary to assure grease circulation within the bearing.

As an alternate we recommend ball bearings of modified 440-C rings and ball material equipped with Rulon retainers to permit dry operation.

This recommendation is based on test experience accumulated with dry bearings, in various atmospheres, over temperature ranges from 350°F to -300°F at various laboratories including NASA, Lewis Research Center, Bureau of Standards Cryogenic Laboratory and M-R-C Research Laboratories. We have repeatedly demonstrated that 204S size beari are capable of dry operation, obtaining lives far in excess of the requirements of thi category.

As a second alternate we recommend ball bearings with rings and balls of modified 440-C, retainers of S-Monel, with retainers and balls dry film coated with CBS CDL-5940 molybdenum disulfide-silver dry film.

Ball bearings with rings and balls of 440-C type material dry film coated with CBS CDL-5940 have been operated for relatively long periods of time (approx. 3000 hrs.) in vacuum tube rotating anodes. In this application the load is low and the speed nominal, for the bearing size involved. In these bearings only the balls were dry film coated. The addition of S-Monel retainers with dry film applied to the retainers as well as the balls, is our recommendation, based on comparative tests of bearings with and without retainers, in air and inert atmospheres.

2.1.4 <u>Oscillating:</u>

For the oscillating bearing category we recognize that aligning ability is a usual requirement for this type of bearing. Such bearings are available in ball and roller antifriction types, as well as plane, swedged or inserted, types.

As our first recommendation we suggest either antifriction type, with rings and balls or rollers manufactured of modified 440-C material, grease lubricated. These bearings can be sealed with integral bearing seals of synthetic rubber or Teflon.

No reliability problems should be encountered with this type, for the two hour service indicated. Even if grease purging occurs, sufficient lubricant will remain on the critical bearing parts to provide ample lubrication. Even if the integral bearing seals are manufactured of synthetic rubber, the short time of exposure will not permit drastic deterioration.

As an alternate we recommend a plane type bearing, self-aligning, in which the load carrying parts are separated by Teflon impregnated fabric. The bearing parts should be manufactured of modified 440-C material.

This recommendation is based on tests at Boeing Airplane Company, evaluating various types of plane spherical rod-end and control bearings (Refs. 69, 113).

2.2 Three-Month Continuous Operation:

In the analysis of expected service duty we have next considered mechanisms which will be required to operate continuously for a three-month mission.

2.2.1 High Load-Low Speed:

For the high load-low speed and low load-high speed categories, it is important that thrust be avoided except to locate parts. This may be accomplished only by careful consideration of the basic mechanism design to make sure that no more than incidental locating thrusts are applied to the bearings.

As our first recommendation in this category we suggest cylindrical roller bearings in which the rings and rollers are manufactured of Stellite 6B. These bearings should be equipped with retainers of Ni-Resist, and the retainers dry filmed with NAMC AML-23A dry film material.

This recommendation is based on experience gained with this material combination and dry film in our work under Contract AF 33(616)-6650 to develop bearing design criteria for 1200°F and 250,000 ft. simulated altitude (Ref. 112). Actually there is very little preference between Stellite 6B and Rene' 41 for this application. Both are corrosion resistant, and both are capable of resisting the friction generated temperatures expected in this bearing.

As an alternate we recommend the plane type bearing with the bearing parts manufactured of Stellite 6B, separated by a liner of Teflon impregnated fabric.

2.2.2 Low Load-High Speed:

In the low load-high speed category we recommend ball bearings, with rings and balls manufactured of Rene' 41 material, and fitted with retainers of Rulon. This bearing will operate dry.

Experience gained with high temperature bearings at M-R-C Research and with cryogenic bearings at the Bureau of Standards Cryogenic Laboratory, and NASA, Lewis Research Center, indicates that this bearing combination is capable of operating at relatively high speeds in the absence of fluid lubrication. The maximum speed is temperature limited, indicated the necessity for applying only low loads and minimizing internal friction.

As an alternate we suggest ball bearings with rings and balls manufactured of modified 440-C material and retainers of S-Monel. The balls and retainers should be coated with CBS CDL-5940 dry film material.

The ability of CBS dry film to lubricate at higher speeds has not been positively established. The ample lives obtained at lower speeds warrant consideration of the bearing configuration as an alternate in this category.

2.2.3 Low Load-Low Speed:

In the low load-low speed category we recommend first a plane type bearing with the bearing parts manufactured of modified 440-C, separated by a liner of impregnated Teflon or Teflon impregnated fabric.

Selection between the two types of Teflon liners will depend on the load level. The Teflon impregnated fabric develops higher friction and suffers greater wear than the impregnated metal. However, the former will withstand unit bearing pressures of only 2000 psi, while the latter has been tested to 50,000 psi (Refs. 46, 49).

In the case of the fabric, difficulties have been encountered with bonding to steel parts, and swedging of prebonded parts.

Solid Teflon liners have exhibited some undesirable characteristics, leading to the development of the impregnated Teflon type of materials.

As an alternate we recommend a dry ball bearing with rings and balls manufactured of modified 440-C and equipped with a Rulon retainer.

This bearing design must be considered a very strong alternate to the preceding recommendation , on the basis of tests run at M-R-C Research and contemporary experience gained at NASA, Lewis Research Center and Pratt and Whitney Aircraft.

As a second alternate we recommend a ball bearing with rings and balls manufactured of modified 440-C equipped with retainers of S-Monel. The retainers and balls should be coated with CBS CDL-5940 dry film.

The conditions of this application closely simulate those under which CBS CDL-5940 dry film has been developed and tested. On this basis a life of 3000 hours may be considered to have been established, although the specific testing conditions can hardly be considered the basis for a positive generalized recommendation.

2.2.4 Oscillating:

In the oscillating bearing category we recommend a self-aligning ball or roller bearing with rings and balls or rollers manufactured of modified 440-C, the use of a Teflon boot equipped with a spring operated pressure relief valve set to hold pressur slightly above the vapor pressure will permit the use of grease lubrication.

We must emphasize the advantages in life and reliability obtained by the use of greas lubrication as compared to dry bearing operation. While we have avoided the recommendation of oil or grease for the thirty-day service requirement in the other bearin applications, it is because we could not envision the use of a hermetic seal to retai the lubricant. With this arrangement, as described above, we strongly recommend the use of grease lubrication. The use of boots has been common practice in aircraft mechanisms for many years, and the only problems which arise in the adaptation of this design to outer space is in the selection of the boot materials and in the development of the spring operated pressure relief valve. These problems certainly should not prove to be insurmountable.

As an alternate we recommend a plane type aligning bearing in which the bearing parts are manufactured of modified 440-C, separated by a liner of impregnated Teflon or Teflon impregnated fabric.

2.3 Intermittent Operation During Two-Year Mission:

We have next considered recommendations for bearings for mechanisms for a two-year mission, in which the mechanisms will be required to operate only for ten to twelve short periods of time; for approximately two-hour cycles.

2.3.1 High Load-Low Speed:

For this duty, and in the high load-low speed category, we recommend cylindrical roller bearings with rings and rollers manufactured of Stellite 6B equipped with retainers of Ni-Resist, with the retainers coated with NAMC AML-23A dry film.

It must be appreciated that dry film lubricated bearings are not capable of sustaining as much load, even at low speed, as oil or grease lubricated bearings. In this case it is estimated that the maximum load for temperature stabilization may not be above that representing a C/P of 3.

As an alternate we recommend a plane type bearing with component parts manufactured of Stellite 6B, separated by a liner of impregnated Teflon or Teflon impregnated fabric.

Available data indicate that liner materials of these types will not deteriorate, even over a two-year exposure time (Ref. 77). On this basis this recommendation appears logical for the same reason and on the same basis as used for its selection for the three-month duty cycle.

2.3.2 Low Load-High Speed:

In the low load-high speed category we recommend ball bearings with rings and balls of Rene' 41 material equipped with Rulon retainers. These bearings are to operate dry.

This is the same recommendation as made for the three-month duty cycle for the low load-high speed category. The validity of this recommendation is based on evidence that two years' exposure will not deteriorate Rulon retainers.

As an alternate we recommend ball bearings with rings and balls manufactured of modified 440-C, and equipped with retainers of S-Monel. The retainers and balls should be coated with CBS CDL-5940 dry film.

There is no indication that the CBS CDL-5940 dry film will deteriorate due to two years' exposure to the environment of outer space. Therefore, the same alternate recommendation appears to be in order here, as for the three-month duty cycle.

2.3.3 Low Load-Low Speed:

In the low load-low speed category we recommend a plane type bearing in which the bearing parts are manufactured of modified 440-C and separated with a liner of impregnated Teflon or Teflon impregnated fabric.

As an alternate we recommend ball bearings with rings and balls manufactured of modified 440-C and equipped with Rulon retainers which will permit dry operation.

This is the same bearing arrangement recommended as an alternate in the low load-high speed category. At lower speed it should be even more reliable.

As a second alternate we recommend ball bearings with rings and balls manufactured of modified 440-C, equipped with retainers of S-Monel. The retainers and balls should be coated with CBS CDL-5940 dry film.

This alternate recommendation is an extension from the alternate recommendation made for this category for a three month service life requirement. This is justified since there is no evidence to the effect that CBS CDL-5940 dry film will deteriorate over a two year exposure to the environment of outer space.

2.3.4 Oscillating:

In the oscillating bearing category we again recommend grease lubrication applied to modified 440-C, self-aligning ball or roller bearing. The bearing should be protected by a boot either of Teflon or stainless steel bellows. In either case the bearing cavity should be equipped with a spring operated pressure relief valve to maintain a pressure slightly above the vapor pressure of the grease.

In a sealed cavity it should be possible to maintain grease lubricant without loss of its lubricating properties. In order to accomplish this, it will be necessary to provide a durable boot. While it is assumed, on the basis of available literature, that Teflon will not deteriorate over a two year exposure, when subjected to flexing, a back-up recommendation of a stainless steel bellows appears to be in order.

As an alternate we recommend a plane type aligning bearing with bearing component parts manufactured of modified 440-C separated by a liner of impregnated Teflon or Teflon impregnated fabric.

2.4 Continuous Operation During Two-Year Mission:

We have next considered a duty cycle consisting of full time operation over a two-year mission. Paucity of available information makes our recommendations for this service life quite hypothetical. Bear in mind that there are relatively few on-earth bearing applications which have run two years, without some attention, such as relubrication.

2.4.1 High Load-Low Speed:

In the high load-low speed category, for this duty cycle, we recommend a cylindrical roller bearing with rings and rollers manufactured of Stellite 6B, equipped with Ni-Resist retainers dry film coated with NAMC AML-23A.

Assuming no deterioration of bearing materials or dry film over a two year exposure, the factors which will cause such bearings to become inoperative are increased clearance and surface roughening due to wear and accumulation of wear debris in critical bearing areas thus preventing motion of the component parts (Ref. 110). Wear is a function of load and number of cycles. Therefore, the load and the speed should be kept as low as possible in all categories, for this service condition. Contact stress to load ratios must be minimized and the bearing design arranged to promote purging of wear debris from the paths of rolling and sliding contact. The absence of air blast or gravity will make it necessary for the bearing parts themselves to push out the wear debris from these critical areas.

2.4.2 Low Load-High Speed:

For the low load-high speed category we recommend ball bearings manufactured with rin and balls of Rene' 41 material equipped with Rulon retainers, which will permit dry operation.

For the life required, the limiting speed of this bearing type and the significance of minimizing load as pointed out in the preceding paragraph may make of this recommendation only an alternate applicable for both the high load-low speed and low load-high speed categories. We feel that heat generation may well prove to be a problem in either of these categories and that a high temperature ring and ball material will be required. However, if the friction is high enough to generate sufficient heat to necessitate the use of high temperature materials, the wear in the Rulon retainer may well be excessive over a two-year service life. In the face of completely inadequate experience and test data we can only recommend this bearing material combination as constituting a likely candidate for this service.

2.4.3 Low Load-Low Speed:

For the low load-low speed application we recommend ball bearings with rings and balls manufactured of modified 440-C equipped with Rulon retainers which will permit dry operation.

If the load and speed are low enough to prevent excessive temperature build-up, modified 440-C should be entirely satisfactory for the rings and balls for this bearing. For reasons listed above, modified 440-C may represent as valid a recommendation as the high temperature materials such as Rene' 41 or Stellite 6B. It is encouraging to note that, all factors being equal, the conditions in this category are less severe than in the preceding two categories and, therefore, there is a better chance of attaining two years' full time operation in this case.

2.4.4 Oscillating:

For the oscillating bearing category we recommend grease lubricated modified 440-C ball or cylindrical roller bearings protected by a Teflon or stainless steel bellows boot, equipped with a spring operated pressure relief valve to maintain a pressure slightly above the vapor pressure of the grease.

We are most optimistic concerning this bearing arrangement. Hermetically sealed grease lubricated bearings have been operated for over two years of continuous service on earth. If the same type of lubrication can be provided, a high degree of reliability should be assured for outer space operation. The life of a Teflon boot under constant flexing against a two-year life requirement may be questionable. In this case the selection of stainless steel bellows type boot is indicated.

TABLE 4

DESIRED LIFE		HIGH LOAD LOW SPEED	LOW LOAD HIGH SPEED	LOW LOAD LOW SPEED	OSCILLATING
2 Hr. Continuous Operation at Start of Mission	Recomm.	Ball or roller bearing Oil lubrication Matl - M-50 Silver plated bronze retainer	Ball or roller bearing Oil lubrication Matl - M-50 Silver plated bronze, Bakelite compound ret.	Ball bearing Grease lubrication Matl - M-50 S-Monel	Self-aligning ball or roller bearing (antifriction) Grease lubrication Matl - 440-C modified
	1st Alt.	Ball or roller bearing Grease lubrication Matl - M-50 S-Monel retainer	Ball or roller bearing Grease lubrication Matl - M-50 S-Monel retainer	Ball bearing (dry operation) Matl - 440-C modified Rulon retainer	Self-aligning plane bearing Matl - 440-C modified Teflon impreg. Teflon fabric
	2nd Alt.	Ball or roller bearing Matl - Rene' 41 Ni-Resist retainer NAMC AML-23A film	Ball or roller bearing Matl - Rene' 41 Ni-Resist retainer NAMC AML-23A film	Ball bearing Matl - 440-C modified S-Monel retainer CBS-CDL 5940 film	
6 Mo. Continuous Operation	Recomm.	Roller bearing (no thrust) Matl - Stellite 6B Ni-Resist retainer NAMC AML-23A film	Ball bearing (no thrust) Matl - Rene' 41 Rulon retainer	Plane bearing Matl - 440-C modified Teflon Impreg. Teflon fabric	Self-aligning ball or roller bearing (antifriction) Matl - 440-C modified Grease lubrication with Teflon boot
	1st Alt.	Plane bearing Matl - Stellite 6B Teflon Impreg. Teflon fabric	Ball bearing Matl - 440-C modified S-Monel retainer CBS-CDL 5940 film	Ball bearing Matl - 440-C modified Rulon retainer	Self-aligning plane bearing Matl - 440-C modified Teflon impreg. Teflon fabric
	2nd Alt.			Ball bearing Matl - 440-C modified S-Monel retainer CBS-CDL 5940 film	
30 Hr. Intermittent Operation	Recomm.	Roller bearing (no thrust) Matl - Stellite 6B Ni-Resist retainer NAMC AML-23A	Ball bearing (no thrust) Matl - Rene' 41 Rulon retainer	Plane bearing Matl - 440-C modified Teflon Impreg. Teflon fabric	Self-aligning ball or roller bearing (antifriction) Matl - 440-C modified Grease lubrication with boot or bellows
	1st Alt.	Plane bearing Matl - Stellite 6B Teflon Impreg. Teflon fabric	Ball bearing Matl - 440-C modified S-Monel retainer CBS-CDL 5940 film	Ball bearing Matl - 440-C modified Rulon retainer	Self-aligning plane bearing Matl - 440-C modified Teflon Impreg. Teflon fabric
	2nd Alt.			Ball bearing Matl - 440-C modified S-Monel retainer CBS-CDL 5940 film	
1 Yr. Continuous Operation	Recomm.	Roller bearing (no thrust) Matl - Stellite 6B Ni-Resist retainer NAMC AML-23A film	Ball bearing (no thrust) Matl - Rene' 41 Rulon retainer	Ball bearing Matl - 440-C modified Rulon retainer	Self-aligning ball or roller bearing (antifriction) Matl - 440-C modified Grease lubrication with boot or bellows

Section C

GEARS

1. DISCUSSION

We have reviewed the gear problems at JPL, and find that the total range of applications covers lightly loaded gears to heavily loaded gears, operating at speeds from high to so slow as to be practically static. Fortunately, in the majority of cases, high speeds combine with low loads and high loads are applied at low speeds. While most problems can be solved with spur gears there is a desire to use worm gears to achieve high ratios with maximum compactness.

Our literature search failed to disclose design data based on valid tests of gears in realistic vacuum. Tests have been run in gaseous atmospheres such as helium, evaluating materials and dry films (Refs. 63, 64). Such tests are attractive since test equipment and procedures are greatly simplified by the substitution of positive pressure gas systems for hard vacuum. While it may be possible to demonstrate that friction and wear in vacuum is proportional or equal to friction and wear in an environment such as helium, no experiments of this nature are known to have been conducted. Therefore, friction and wear values obtained in gas and even the order of rating materials and lubricants by tests in gas are subject to question.

We feel that the sliding component of friction constitutes a much greater lubricant requirement than rolling contact. Conventional gears induce sliding in the power transmitting contact varying the degree from a minimum in spur gears to a maximum in cross axes, helical gears and worm gears.

While worm gears are especially attractive to obtain high reduction ratio in a compact configuration, a practical design, which will eliminate or minimize the inherent sliding contact for continuous rotation, is difficult to conceive. If only angular rotation of the output shaft is required, a sector of the worm gear can be replaced by a toothed roller supported on antifriction bearings. This design, which is used in automotive steering mechanisms, greatly reduces the component of sliding motion between the worm and worm gear sector.

Gears are more difficult to lubricate than bearings or seals because no suitably proportioned surfaces appear to retain dry film or other friction reducing materials. In this connection, experiments have been run with idler gears of friction reducing materials in an attempt to continuously replenish a friction reducing coating on the load carrying gears. This ingenious arrangement has not proven universally satisfactory especially for high speed gears.

Our recommendations for immediate design decisions involve the use of fluid lubrication for every application in which the retention of lubricant is practical. For those applications in which it is not possible to introduce fluid lubrication, it is necessary to sacrifice load carrying capacity and reliability by the introduction of dry film lubricants, unless the sliding action of gears can be replaced by rolling action such as is possible with roller chains or rocking pin link tooth chains.

We recognize that the replacement of gears by chains must be done at time of basic layout of the mechanism and that serious design complications may arise in the course of the application of this design concept.

In our research suggestions to follow, we discuss some alternate design concepts which should be considered for further development. At this time, however, they are considered to be too experimental in nature to be considered for current designs.

Please bear in mind that the recommendations which follow are based on our interpretation of experience with gears to date, tempered by our knowledge of materials and friction reducing agents, which knowledge has been obtained primarily with antifriction bearings. These recommendations should be considered as tentative. We recognize the need for further research and development to determine load-life-speed relationship and assure reliability in contemporary design. Much less attention has been directed to the problem of operating gears in space than has been directed to bearings. We also recognize, however, the need for design suggestions and recommendations now.

2. GEAR APPLICATIONS (See Table 5)

On the basis of our study of gears to operate in outer space, we divide the problem into four areas involving spur gears, straight bevel gears, and worm gears including spiral bevels and hypoids, as well as nonparallel shaft helical gears. In the first category we have further divided spur gears into the high load-low speed and low load-high speed categories.

In this analysis various materials and lubricants are discussed. These are listed, by composition, elsewhere in this report. In the case of gear materials, the design of the gear part frequently dictates the material selection. For example, for some shapes, surface hardened material is preferred to through hardened material.

As far as the lubrication requirements are concerned, the following materials are interchangeable:

a. M-50 (through hardened) consumable electrode vacuum melted

b. Modified M-50 (carburized) consumable electrode vacuum melted

c. H-12 (nitrided) consumable electrode vacuum melted

NOTE: Alloy content of materials is shown in Table 2.

To date, development of the CBS CDL-5940 dry film has been directed toward lubricating type 440-C stainless steel materials. Pending further work on other contact materials, we must recommend 440-C or modified 440-C (440-CM), with CBS CDL-5940 dry film.

2.1 Two Hours Continuous Operation at Start of Mission:

For mechanisms required to operate only at the beginning of a mission, for approximately two hours, we strongly recommend oil lubrication for all types of gears. There are various candidate oils for this application.

Rocketdyne has used Sinclair L-743, a MIL-L-25336 oil with defoaming agent. On the basis of their satisfactory experience, we recommend this oil for all of the various gear types. In bearing tests at MRC, polyphenylether lubricant has demonstrated exceptional stability and has produced extended bearing fatigue lives under extreme operating conditions. These fluids possess low vapor pressures. Polyphenylether lubricant is produced by Monsanto, as OS-124.

Consideration should be given, also, to a new class of fluids, known as fluorosilicones. On the basis of Shell Four-Ball tests, they are superior to petroleums, silicones, and polyphenylethers. Such fluid is produced by Dow Corning, as FS-1265. Differing oil characteristics may be desirable for the different types of gears. For example, an E. P. type additive may be of advantage in the lubrication of worm gears, spiral bevels, hypoids, and nonparallel helical gears.

As first alternate to oil lubrication we recommend grease lubrication for all types of gears indicated. In this connection the grease cavities should be so designed as to provide in-feeding of lubricant into the critical gear areas. This is especially important in the case of low speed gears in which adequate agitation may not be available to restore grease to the teeth after it has been squeezed out by meshing

Success in the lubrication of gears in normal applications has long been felt to be a function of the viscosity of the gear lubricant. While some correlation exists with oil viscosity as measured at atmospheric pressure, it is felt that actually the viscosity of the oil film under contact pressure is the more logical criterion.

The oil lubrication of gears in space application even where relatively poor sealing is to be provided, does not appear to create a serious problem. An ample supply of gear oil, even though high evaporation may be encountered, should provide adequate lubrication. Tests have shown that in inert atmospheres, such as Argon and Nitrogen, the load carrying capacity of gear oils is greatly increased (Ref. 100). It may be that in the absence of any atmosphere a similar improvement will also be noted.

The following products are recommended for the oil lubrication of gears:

Manufacturer	Product	Type
Dow Corning	XF-258	Phenyl-methyl-polysiloxane
General Elec.	F-50	Silicone
Sinclair	L-743	High film strength ester (MIL-L-25336)
Shell Oil	Aeroshell Fluid 5L	Mineral Oil

The grease lubrication of [illegible]rs, especially at high speeds, creates somewhat more of a problem. At high speeds and high loads the necessity [illegible]r adequate film strength becomes apparent. Gear teeth, regardless of their design, or the accuracy with which they are made, develop a combination of sliding and rolling motion as they pass into and out of mesh. Since sliding always prevails between gear teeth, accompanied so often by very high pressures which are comparable to those encountered in rolling contact bearings, maintenance of the lubricating film is an important requirement. If grease is to be used it must be soft enough so as to transfer good adhering lubricant films to the gear teeth. The oil used in compounding such a grease should have proven to be effective in gear lubrication.

The following greases are suggested as possible lubricants for gears:

Manufacturer	Product	Type
Shell Oil	Aeroshell 7A	EP Synthetic (MIL-G-7118A)
General Elec.	Versilube G-300	Soap thickened F-50 fluid
Dow Corning	FS-1291	Soap thickened fluorosilicone

For both categories of spur gears, we recommend a second alternate of M-50 high speed steel gears lubricated with NAMC AML-23A dry film. This combination has been proven for varying loads and speeds, providing life adequate for this environmental requirement.

For lightly loaded straight bevel, helical, worm and spiral bevel, including hypoid gears, we suggest a second alternate of Teflon impregnated metal, operating without additional lubrication. For this material, the allowable tooth load is approximate 5% of the allowable for lubricated steel.

The inherent poor strength and poor resistance to cold flow of Teflon has been overcome by the use of Teflon to impregnate porous solids. One such product is Sinitex produced by Booker-Cooper, Inc., North Hollywood, California. This material is essentially a sintered bronze-tin (55%) containing Teflon (27%) and molybdenum disulfide (18%). This combination provides a low friction bearing material with goo strength properties and should be capable of generating Teflon-MoS_2 films.

This approach appears to be superior to the use of Teflon film coatings, applied from dispersions, where the quality of the bond may be questionable. The use of

Teflon auxiliary idler gears appears to be promising for heavily loaded applications. Some difficulty, requiring ingenious design to overcome, may be encountered to obtain tne proper tooth contact to assure a sustained lubricant film.

2.2 Three-Month Continuous Operation:

The next application to be considered involves full-time operation of gears during a 30-day mission. For this duration we have eliminated fluid lubrication from consideration.

For the high load-low speed application, we recommend that gears be avoided, and that use of a chain be considered. Both roller chains and rocking action timing chains appear promising for this application.

In spur gears, pure rolling occurs only at the pitch line. Over the remaining contact increments, sliding increases friction and wear, deteriorates the finish and causes increasing friction heat rejection. As this process continues, vibration and shock loading increase, causing a stress increase at the tooth root. Thus, tooth wear can cause tooth breakage.

It is difficult to retain lubricant, fluid, grease or dry film, on the gear teeth. Loss of lubricant accelerates the wear process noted above. At high load level, this sequence becomes precipitous.

In a roller chain, rolling action decreases the incremental sliding between chain roller and sprocket. Up to critical load and speed values, this action is predominately rocking, with only a minimum amount of sliding occurring, between the roller and its bushing. This contact not only operates at reduced stress, but is of such a geometry as to facilitate the retention of lubricant.

As a further development, directed toward maintaining rocking action in the chain hinge, the Morse Chain Company has developed the HiVo chain and proven this design capable of higher speed and power ratings than conventional timing chains and roll chains. This chain type has been operated over long service lives lubricated with molybdenum disulfide applied at time of manufacture.

For the low load-high speed application, we recommend the use of spur gears, incorporating Teflon as a lubricating agent. One such method of utilizing Teflon appea promising for this application, as previously described.

As an alternate we suggest the use of M-50 high speed steel gears lubricated with NAMC AML-23A dry film.

This same recommendation and alternate applies for straight bevel gears. Consideration has been given to the sliding component of the tooth contact in this gear type. Wear rates will be greater, and allowable tooth loads lower, than for spur gears.

We have no recommendation to offer for materials or lubricants for worm gears, spiral bevels, hypoids or nonparallel shaft helical gears for this service requirement. We can only suggest that this gear type be avoided by the use of chains and spur or straight bevel gears. Planetary sets may be used to accomplish high reduction ratios.

2.3 Intermittent Operation During Two-Year Mission:

For applications involving a two-year mission, in which the gears are required to operate intermittently for only short periods of time, we have the following recommendations to offer:

For the high load-low speed application we suggest again that spur gears be avoide substituting instead chains of either the roller or the rocking pin timing chain type.

For the low load-high speed spur gear application we suggest the use of gears manufactured of Teflon impregnated material. As an alternate we again suggest the use of M-50 steel gears lubricated with NAMC AML-23A dry film.

The same recommendations apply for straight bevel gears in the order listed for the low load-high speed spur gear application.

For worm gears, spiral bevels and nonparallel shaft helical gears, we recommend the use of modified 440-C materials and CBS CDL-5940 silver-molybdenum disulfide dry film. This combination may be load limited, although the CBS multilayer dry film should provide a better bond to the parent material and, therefore, avoid peeling under the shearing stresses involved in this type of gear contact.

As an alternate we suggest the use of sintered metallic material for the gears, impregnated with Teflon.

2.4 Two-Year Continuous Operation:

For the application involving a two-year mission, with the gears operating 100% of the time, we have the following recommendations to offer:

Again we recommend that spur gears be avoided for the high load-low speed application and that instead, chains, either roller or rocking pin tooth chains be substituted.

For the low load-high speed application for this environment, we recommend the same configuration: a chain drive. We realize that this arrangement may be speed limited.

As an alternate we suggest that the gears be manufactured of metallic material impregnated with Teflon.

For straight bevel gears in this environment we suggest that the gears be manufactured of metallic materials impregnated with Teflon.

In view of the long service life involved, we have no recommendations to offer for materials or lubricants for worm gears, spiral bevels, hypoids or nonparallel shaft helical gears.

TABLE 5

GEAR APPLICATIONS		SPUR GEARS		STRAIGHT BEVEL GEARS PARALLEL SHAFT HELICAL GEARS	WORM GEARS SPIRAL BEVEL GEARS HYPOID GEARS NON-PARALLEL SHAFT HELICAL GEARS
		High Load-Low Speed	Low Load-High Speed		
2 Hr. Continuous Operation at Start of Mission	Recomm.	Oil lubrication* Steel gears*	Oil lubrication Steel gears	Oil lubrication Steel gears	Oil lubrication Steel gears
	1st Alt.	Grease lubrication Steel gears	Grease lubrication Steel gears	Grease lubrication Steel gears	Grease Lubrication Steel gears
	2nd Alt.	Dry film lubrication Steel gears	Dry film lubrication Steel gears	No lubricant Teflon filled metal gears	No lubricant Teflon filled metal gears
3 Mo. Continuous Operation	Recomm.	Dry film lubrication applied to roller or HiVo chain, eliminating gears	No lubricant Teflon filled metal gears	No lubricant Teflon filled metal gears	No recommendation - Avoid if possible
	Alt.		Dry film lubricant Steel gears	Dry film lubricant Steel gears	
20 Hr. Intermittent Operation 2 Yr. Mission	Recomm.	Dry film lubrication applied to roller or HiVo chain, eliminating gears	No lubricant Teflon filled metal gears	No lubricant Teflon filled metal gears	Dry film lubricant 440-CM steel gears
	Alt.		Dry film lubricant Steel gears	Dry film lubricant Steel gears	No lubricant Teflon filled metal gears
2 Yr. Continuous Operation	Recomm.	Dry film lubrication applied to roller or HiVo chain, eliminating gears	Dry film lubrication applied to roller or HiVo chain, eliminating gears	No lubricant Teflon filled metal gears	No recommendation - Avoid if possible
	Alt.		No lubricant Teflon filled metal gears		

*See text for specific material recommendations

Section D

SEALS

1. DISCUSSION

Through the years two basic types of rotary shaft seals have evolved which provide very low leakage. These are the packing gland and the face type mechanical seal. A third type which has received some attention is the centrifugal seal in which the centrifugal force of a liquid sealant resists flow of the fluid being sealed.

The sealing methods now being used at JPL are of the packing gland type. Two O-rings are used about the shaft with an annulus space between, filled with the lubricant used within the chamber.

As far as is known, no rotary seals are presently available commercially for high speed transmission through a vacuum wall. In a program being carried out at Goodrich High Voltage Astronautics, Inc., Burlington, Massachusetts, investigations include face type seals, metallic lip type seals and liquid metal sealing media.

A seal unit using synthetic rubber lip seals has accumulated 50 hours of running time at vacuums between 10^{-5} and 10^{-6} TORR at 3600 RPM. A low vapor pressure oil contained in the chamber lubricates the seal surfaces. This type of seal is limited to rubbing speed between 1500 and 3000 feet per minute.

Tests have also been conducted on graphitic carbon face seals at speeds ranging from 2000 to 10,000 RPM at vacuums of 2×10^{-5} TORR or better, with a system pressure of 10^{-5} TORR. While test durations to date have been relatively short, this type of seal appears attractive for relatively high speed shafts. A small quantity of oil is required to lubricate the stationary carbon face and rotating the steel mating ring.

GHVA is also investigating a new metallic lip seal, similar in construction to the standard synthetic rubber lip seal, except that it is made of steel (Ref. 75).

This report goes on to state:

> "It is worth noting that the seal requirements on a vacuum system which has a limited pumping speed, are more rigorous than those in space which has an infinite pumping speed. In space, the primary aim would be to minimize loss of lubricant or fluid and prevent contamination of the part. The problem of seals and bearings for use in space still exists, but encouraging results have been obtained and there remain several promising avenues of investigation."

1.1 Contact Seals:

Despite the fact that minimum leakage contact seals constitute sliding contact with asperity cleavage, there is a strong possibility that PTFE or even dry films may permit the development of positive contact seals in which the contact pressures are so low and closely controlled that long life may be obtained. It is suggested that these seals will be of the axial type rather than the radial type, since it is much easier to control contact pressures by axial deflection of a bellows or diaphragm, than to maintain relatively constant contact pressure over the small available radi displacement of a continuous annular member (Ref. 42).

1.2 Centrifugal Seals:

Several years ago Battelle Memorial Institute performed a considerable amount of research on low leakage centrifugal seals for vacuum systems (Ref. 86). By using a simple rotating annulus and stationary vane design and using mercury as the sealant, it was found that leakage was excessive. The lowest absolute pressure obtainable was 5.5×10^{-3} mm Hg. Analysis indicated that turbulence created at the boundary of the stator was the cause for the high leakage rate - approximately 160 cubic inches of standard air per year. The state of the art was advanced by placing baffles in the rotating annulus such as to minimize sealant turbulence at the stator boundary and leakage was reduced to 22 cubic inches of standard air per year. This type of seal design would appear to have the capability of providing very low leakage rates for sealing against vapor and since it generates its own centrifugal sealing pressure, it would also function in a zero G environment. This type of configuration is covered in U. S. Patent 2,606,044 (Ref. 70).

Other types of centrifugal seals show promise for this type of application, including simple flingers which would also have the advantage of dispersing fluid creepage along the shaft.

1.3 Labyrinth Seals:

Labyrinth seals have been suggested for minimizing the escape of particles outgassing from a body of lubricant, such as a low vapor pressure fluid. In this concept it is proposed that mirror-like surfaces be incorporated in the baffle convolutions, thus reflecting back any molecular efflux.

2. SEAL APPLICATIONS (See Table 6)

Investigation of this subject indicates that seals can be divided into two basic categories: those for rotational applications and those intended for sealing shafts which do not rotate one full turn and which may be termed as oscillating. We assume that seals are required, other than to retain lubricants, in some applications.

Within the rotational category, seals may be divided into the high speed and low speed categories although it is difficult to establish a definite line of demarcation between the two.

Again we have considered four different mission requirements.

2.1 Two-Hour Continuous Operation at Start of Mission:

As shown in the tabulation, we believe that, for high speed rotational seals, the face seal design must be incorporated. There are three alternative material and lubricant selections in this category:

a. With oil or grease lubrication, either a graphite or Teflon rubbing member may be used in combination with a hardened steel seal plate.

b. A Teflon seal member may be operated dry, in contact with a chrome plated steel plate. As in the case with plane bearing operation, the question of heat conduction during high speed operation must be considered. In recommending the use of Teflon for one seal member, it is assumed that DU metal, metal impregnated with Teflon (see Discussion, Section C, GEARS) or other combinations of Teflon and heat conducting reinforcement will be applicable to high speed seal design.

c. It is indicated that NAMC AML-23A molybdenum disulfide type dry film is capable of lubricating a face seal in the absence of any other type of lubrication. We feel that the performance of this dry film can be optimized by the use of a Stellite 6B seal member rubbing against a CA-3 tungsten carbide seal plate.

In the low speed rotational category, we feel that a conventional lip seal should be adequate with oil or grease lubrication. Over the temperature range indicated, either a rubber or a Teflon elastomer material may be used, in contact with a smooth, As an alternate, for dry operation, the same Teflon type seal member, contacting a chrome plated steel seal plate, may be used, as is suggested for the high speed application.

Similarly, the dry film lubricated face seal arrangement, incorporating a Stellite 6B seal ring against a tungsten carbide seal plate, must be considered in this category.

For the short service duration involved, a lip seal may be used in the oscillating application involving oil or grease lubrication. Again, a rubber or Teflon lip seal material may be incorporated. In this application, a spring loaded pressure relief valve should be incorporated, so designed as to bring the pressure in the sealed cavity to a level slightly above the vapor pressure of the fluid to be sealed.

Interchangeable with this concept, consideration should be given to the incorporation of a boot, providing a hermetic seal by the incorporation of a rubber or Teflon member which will retain oil or grease lubrication within the bearing. Again, a spring loaded pressure relief valve should be incorporated to bring the cavity pressure to the vapor pressure of the fluid to be sealed.

2.2 Three-Month Continuous Operation:

For applications to endure a 90-day mission, in which the mechanism is required to operate full time during this period, consideration of fluid lubrication must be eliminated.

In the case of the high speed rotational seal application, the first suggestion is the face seal operating dry with a seal ring of Teflon, in contact with a chrome plated steel seal plate.

The alternate arrangement is, again, the NAMC AML-23A dry film lubricating a Stellite 6B seal ring in contact with a tungsten carbide seal plate.

As another alternate we suggest consideration of a centrifugal seal similar to that described in U. S. Patent 2,606,044. In this design a rotating annulus is partially filled with a liquid and a stationary vane dips into the liquid. Centrifugal force effects a seal closure. For application to the present problem there is a need for a high density, low vapor pressure sealant which will remain liquid over the indicated temperature range. This type of seal design appears to have the capability of providing very low leakage rates and since it generates its own centrifugal sealing pressure, it would also function in a zero G environment.

The first two designs suggested for the high speed rotational seal application also apply for the low speed application. These are:

1. The Teflon seal ring in contact with a chrome plated steel seal plate.
2. The Stellite 6B seal ring contacting a tungsten carbide seal plate, lubricated with NAMC AML-23A molybdenum disulfide type dry film.

The centrifugal seal does not appear applicable for this low speed application.

For the oscillating application, even for 90-day missions, oil or grease appears practical, contained in a Teflon boot. A spring operated relief valve should be incorporated, adjusted to a pressure slightly above the vapor pressure of the lubricant contained.

As an alternate, the boot design can be accomplished in a metallic material. For example, an 18-8 stainless steel bellows has been designed to permit as much as 360° rotation of an arm and shaft combination. With this design, oil or grease lubrication can be used. Again, a spring loaded relief valve should be incorporated, adjusted to slightly over the vapor pressure of the contained fluid.

2.3 Intermittent Operation During Two-Year Mission:

We next consider the requirements of seals for a two-year mission, in which the mechanism will operate for only short periods of time of approximately two hours each.

For the high speed rotational application, it would appear that the same designs suggested for the previous application conditions should be applicable.

A face seal design to operate dry, utilizing a seal ring of Teflon and a seal plate of chrome plated steel, should represent the most reliable recommendation to be made at this time.

As an alternate, we suggest the use of a Stellite 6B seal ring in contact with a tungsten carbide seal plate, lubricated with NAMC AML-23A dry film.

We have avoided the recommendation of the centrifugal seal in this case, due to the long periods of nonrotation.

The low speed designs to be suggested are the same as for the high speed, in this category.

For the oscillating application, in view of the two-year exposure, a metal boot may be preferable to an elastomer, even Teflon. Oil or grease lubrication can thus be incorporated, together with a spring operated relief valve holding the pressure slightly above the vapor pressure of the fluid sealed.

2.4 Two-Year Continuous Operation:

The last application category involves a two-year mission, in which the mechanism is required to operate 100% of the time.

For the high speed rotational seal, the centrifugal sealing arrangement now becomes the most promising, with an alternate of the Stellite 6B seal ring contacting a tungsten carbide seal plate lubricated with NAMC AML-23A dry film.

For the low speed rotational application, the most promising design incorporates a face seal operating dry, with a Teflon seal ring, contacting a chrome plated steel seal plate.

As an alternate, consideration should be given to the Stellite 6B seal ring contacting a tungsten carbide seal plate, lubricated with NAMC AML-23A dry film.

For the oscillating application, the metal boot again appears most promising, allowing oil or grease lubrication. This design, again, should incorporate a spring loaded relief valve, holding the pressure slightly over the vapor pressure of the fluid to be sealed.

TABLE 6

SEAL APPLICATIONS		ROTATIONAL - HIGH SPEED	ROTATIONAL - LOW SPEED	OSCILLATING
2 Hr. Continuous Operation at Start of Mission	Recomm.	Face Seal Oil or grease lubrication Graphite or Teflon vs. steel	Lip Seal (O-ring) Oil or grease lubrication Rubber or Teflon vs. steel	Lip Seal Oil or grease lubricati Rubber or Teflon vs. st Pressure relief valv
	1st Alt.	Face Seal Dry Teflon vs. chrome plate on steel	Face Seal Dry Teflon vs. chrome plate on steel	Boot Oil or grease lubricati Rubber or Teflon Pressure relief valve
	2nd Alt.	Face Seal Dry Stellite 6B vs. Tungsten Carbide NAMC AML-23A film	Face Seal Dry Stellite 6B vs. Tungsten Carbide NAMC AML-23A film	
3 Mo. Continuous Operation	Recomm.	Face Seal Dry Teflon vs. chrome plate on steel	Face Seal Dry Teflon vs. chrome plate on steel	Boot Oil or grease lubricati Teflon Pressure relief valve
	1st Alt.	Face Seal Dry Stellite 6B vs. Tungsten Carbide NAMC AML-23A film	Face Seal Dry Stellite 6B vs. Tungsten Carbide NAMC AML-23A film	Boot or bellows Oil or grease lubricati Stainless steel Pressure relief valve
	2nd Alt.	Centrifugal Seal (Apiezon K Fluid)		
20 Hr. Intermittent Operation 2 Yr. Mission	Recomm.	Face Seal Dry Teflon vs. chrome plate on steel	Face Seal Dry Teflon vs. chrome plate on steel	Boot or bellows Oil or grease lubricati Stainless steel Pressure relief valve
	Alt.	Face Seal Dry Stellite 6B vs. Tungsten Carbide NAMC AML-23A film	Face Seal Dry Stellite 6B vs. Tungsten Carbide NAMC AML-23A film	
2 Yr. Continuous Operation	Recomm.	Centrifugal Seal	Face Seal Dry Teflon vs. chrome plate on steel	Boot or bellows Oil or grease lubricati Stainless steel Pressure relief valve
	Alt.	Face Seal Dry Stellite 6B vs. Tungsten Carbide	Face Seal Dry Stellite 6B vs. Tungsten Carbide	

Section E

RESEARCH PROGRAMS

1. INTRODUCTION

In the design of lubricating systems for spacecraft mechanisms, it has been reasonable, to date, to attempt to use conventional materials and lubricants by the control of environment adjacent to the bearing area. While the problems peculiar to space have been recognized for a number of years, there has been more talk than action and new approaches have not been developed to meet them. In order to permit space programs to move forward, it has been necessary to extrapolate from existing knowledge, and in some cases, compromise the design of spacecraft to permit the use of existing technology.

While initial interest in space exploration caused considerable concern over the demandwhich vacuum, radiation, and prolonged exposure time would impose on materials and lubricants, there is now the possible danger that apparently successful operation of space mechanisms, employing relatively simple lubrication techniques, will tend to mitigate the seriousness of the need for research and development effort in this area.

Discussion of friction research and development raises many conflicting opinions. There is, at one extreme, an expression that frictional problems are so complex and the variables so numerous that anyting short of an all-out, crash program to develop radically new concepts is useless. Others feel that the problem is much over rated and may be solved, on the basis of present activity, by "fix-it" measures during development. In general, the depth and breadth of the subject is not fully appreciated by the scientist, to whom lubrication is secondary to their field of specialty, the designer who must incorporate lubrication techniques, and the lubrication specialist who is sales oriented or experimenting. The problem lacks concise definition, in terms common to all concerned.

There is need to question the role which contractors, assigned to develop specific hardware, will play in establishing such basic specifications as the bearing materia and lubricant to be used. For example, a contractor, selected for his skill in electronics, may not have the experience nor the personnel to cope with the lubrication problems which are attendant to his over-all responsibility.

MRC has recently experienced a case in point. One contractor was assigned responsibility to design a large, multi-million dollar system. As their subcontractor, we conducted experimentation and developed specifications for bearings, materials, and lubricant to meet their requirements. A second contractor, assigned to construct the system, chose to ignore this experimental work, revised the criteria for lubricant selection, and required another experimental program to be run. An appreciable loss of time and money resulted from lack of coordination.

1.1 <u>The Need for Research:</u>

There is a definite need, within the scope of JPL activity, for applied research and development effort in several areas. There is such a variety of lubrication problem to be met that it is inconceivable that a single approach can most satisfactorily meet requirements of all of them. The charts of recommendations which appear elsewhere in this report serve to illustrate the scope of the problem. Admittedly, these charts involve much guesswork and serve to raise many questions. It is to the answer of these questions that research must be directed.

It was intended, originally, that this report should only cover bearings. There is, however, sufficient similarity in the basic frictional problem of bearings, gears, seals and electrical contacts and in the application of these machine elements to various mechanisms, to make a coordinated research program in all these areas feasible and potentially productive. To accomplish this, certain prerequisites should be met:

A. At the present time, lubrication is a part-time function of the Materials Group at JPL. It deserves more emphasis. A full-time lubrication specialist should be given sufficient personnel and experimental facilities to support lubrication development for in-house projects. It is our understanding that, although Prospector and Surveyor will be subcontracted, other projects following Ranger will be retained in house.

B. All contracted research programs involving the study of frictional problems in bearings, gears, seals and electrical contacts should be coordinated through a single group. This may be an outside research organization.

C. Coordination of research activity should include the formation of an advisory group from outside JPL to facilitate communication with activity of other research agencies and across the many disciplines that will be involved.

1.2 Coordination of Research Activity:

1.2.1 Environmental Simulation:

Before embarking on any research program, attention should be directed to the problem of test equipment and procedures. There is need to standardize the requirements for control of environmental variables in experimental programs. Specifications for test conditions and procedures should be coordinated beforehand and not left to the discretion of the individual contractor. There are several references to this problem (40, 54, 57, 60, 67, 71, 72), which cause the value of much of the work which has been done to date to be questioned. The following discussion is drawn from these reports:

To accurately simulate the vacuum of space, consideration must be given to the long mean free path of molecules and atoms. In the relatively close confine of a vacuum chamber, absorbed vapors, molecules and atoms leaving the bearing surfaces are deflected off the walls of the chamber and redeposited or reabsorbed. Even the most minute quantities of these contaminating vapors can have an appreciable effect on friction and wear phenomena. In space, the same desorbed molecules have little chance of returning to the original surface and are quickly lost. While it is evident that actual space environment cannot be duplicated, care should be taken to control molecular contamination.

JPL Technical Report 32-150 suggests that, in order to simulate space vacuum properl in laboratory studies of friction and lubrication, pressures of 10^{-9} TORR or lower should be used. "In no case to date has this been done. This pressure is near the limit of the present state of high vacuum art under conditions where the specimen of interest may be evolving gases and is commonly of a composition which cannot be baked out at high temperatures to clean it."

Because gas evolution plays an important part in the behavior of materials in vacuum it is also suggested that it is important to provide pumps with high pumping speeds at appropriate pressures, as well as the ability to produce low ultimate pressure (Ref. 9).

In ASLE Preprint No. 61-LC-2, Messrs. Buckley, Swikert and Johnson report on the difficulties of finding effective trapping devices to stop back migration of diffusion pump oils. Surfaces of friction test specimens were wetted with distilled water to determine any evidence of oil contamination on the surface. A survey was made of cold trap and baffle designs reported in the literature and employed by users of oil diffusion pumps. At least ten different configurations were tried, none of which completely succeeded in eliminating back migration of vapors from the oil diffusion pump. They finally replaced the oil diffusion pump with an ionization pump which employs no fluid or vapors in its operation.

Correspondence with Dr. Earl G. Jackson of National Research Corporation emphasizes a distinction between back streaming and back diffusion. Back streaming strictly refers to the line of sight transmission of oil molecules from the top of the pump. Dr. Jackson feels that this can be reduced sharply by appropriate optical baffling.

He feels the problem of most concern, and that which is involved in the ASLE paper, is more properly called back diffusion. In this phenomenon, simple optical baffling in the ordinary sense is not adequate because molecules reside for a time on the wall and then volatilize off at random times and in random directions. Such a mechanism can penetrate optical baffling because the angles of incidence and reflection bear no relation to each other. However, such a mechanism should be susceptible to cryogenic baffling by means of which a molecule is completely immobilized once it hits a cryogenic wall. It also appears that various adsorbants, such as charcoal or Zeolites, when maintained at very low temperatures will accomplish the same end.

It is reported that Mr. Paul Bowen at Westinghouse Electric Corporation has found Zeolite to be effective in trapping back diffusion even when using the sensitive wetability detection method outlined in the ASLE paper. This tends to confirm the belief that diffusion pumps can be adequately baffled, although the problem is far more difficult than was at first thought.

Consideration should also be given to the fact that while tungsten-titanium and vac-ion pumps will eliminate the oil migration problem, these pumps are easily choked and are not practical for extended use.

The paper by Hablanian and Steinherz, given at the AVS Symposium, October 1961, points up the relationship between back streaming and inlet pressure. There is a strong possibility that most of the contamination detected in reasonably well baffled systems occurs during the pump-down period and that very little, if any, contamination results during operation at very low pressures. They conclude that back streaming measurements can only be performed after equilibrium conditions are attained. Back streaming rate is a steep function of distance from the mouth of the pump and the inlet pressure. If back streaming must be minimized, the diffusion pump should not be operated at inlet pressures above 10^{-3} TORR.

"Most measurements of back streaming rate are generally conducted at the blank off operation of the pump. One of the important questions is what happens during operation at higher inlet pressure and particularly during start up of the pump. Measurement of the back streaming rate at various inlet pressures were made with a 32 inch pump (HS 32-32,000) ....the rate does not change significantly as long as the inlet pressure is below about 10^{-3} TORR. This is the point where most diffusion pumps have an abrupt reduction in speed indicating that the top jet essentially stops pumping.

"The sudden increase of back streaming at inlet pressures above 10^{-3} TORR points out that a diffusion pump should not be operated in this range unless the condition lasts a very short time. It is common practice in vacuum system operation to open the high vacuum valve after the chamber has been rough pumped to about 10^{-1} TORR exposing the diffusion pump to this pressure. In well proportioned systems this condition lasts only a few seconds, otherwise extended operation at high pressu can direct unacceptably high amounts of oil into the vacuum system."

This same report indicates that new oil diffusion pumps can produce ultimate pressures of about 1 x 10^{-9} TORR, without any baffles or traps. Using Zeolite traps, pressures in the range of 10^{-11} can be reached (Ref. 72).

Our discussions with National Research Corporation of Newton Highlands, Massachusetts have resulted in a proposal for an ultra high vacuum test unit capable of 1 x 10^{-9} TORR in a test volume 14 inches in diameter by 30 inches high. This system utilizes a three pump oil diffusion system with special traps. The equipment incorporates a 250°C bake-out cycle, a cold cap fitted to the top jet of the diffusion pump, a water cooled plate type optical baffle immediately above the pump inlet, and a circular chevron liquid nitrogen cooled trap in the pumping elbow near to the chamber inlet. It is claimed that tests at NRC Laboratories on similar equipment have shown that back migration of oil is positively eliminated. In this system vacuum measurements are made in the top of the chamber with a No. 751 Nottingham Magnetron Ionization Gage.

It is apparent that many experimenters have been careless not only in the equipment used to simulate space vacuum conditions, but also in the instrumentation used to measure conditions and the test procedures employed in experiments. Before embarking on any research program, care should be exercised to determine that laboratory equipment will be able to provide test results which are meaningful to space application.

The state of the art of vacuum technology has advanced considerably just in the past year. It appeas feasible to begin to develop standards for vacuum system components for friction studies. This should be considered as a part of any research effort.

1.2.2 Communication:

Communication of technical data is a profound problem which exists in many technical fields. Even within the field of lubrication, the solution (if indeed there is one) probably rests at a higher national level, such as NASA or Department of Defense. The Defense Metals Information Center at Battelle Memorial Institute is one approach to this problem.

Steps must be taken to avoid duplication of other present effort and to draw on the work being done in other technical fields. The communication problem is twofold: it involves not only surveillance of other agencies engaged in space research, but also recognition of effort in other technical disciplines which may be pertinent to the friction problem. The formation of an advisory committee is, we believe, the best step which JPL can take in this direction.

The scope of JPL activity in the field of frictional research will be a major effort involving several research groups. It is not advisable for a single research institution to undertake the entire project. For the effort to be of maximum usefulness, it is essential that it be coordinated by a single group and related to the specific hardware development within JPL responsibility. We believe the recommendations listed in paragraph 1.1 are essential for this purpose (Ref. 44).

2. NATURE OF THE PROPOSED RESEARCH PROGRAM

Historically, there is a dilemma which has been faced with each new field of technology and has been experienced most recently with respect to nuclear reactors and jet engines. Effort is usually expended in one of two extremes: very basic "university type" research to acquire new knowledge of fundamental phenomena and very applied experimentation directed at successful type tests of a particular system. The middle ground where research is applied to a particular class of hardware, usually suffers.

Apparently this situation now exists with NASA. We hold Contract NASw-72, which was one of the first to be granted when this organization was formed. It concerns the effect of ball fabrication techniques on rolling fatigue life. While this contract has been extended and work continues, we have been told that it does not really fit within the NASA program scope, since, on the one hand, it is not sufficiently fundamental, while, on the other hand, it does not fit within a system program.

In charting the course of friction research for JPL, the most important consideration is the time schedule for space exploration. It is apparent that lubrication problems require a fundamental program to provide designers with solutions before they must be committed. Research work done in parallel with design developments is too late. Research done as part of development is usually not sufficiently general to provide the basis for future design activity. We must avoid forcing each contractor to study and solve each applied problem himself. Basic studies must provide design criteria, before the need.

There are three areas of investigation which should be incorporated in a JPL research program and coordinated over the areas of bearings, gears, seals and electrical contacts:

A. Implement immediate design decisions for application of these machine elements by environmental tests of existing off-the-shelf hardware utilizing existing material and lubricant candidates.

B. Initiate investigation into new mechanical design concepts which will contribute to solution of the lubrication problem.

C. Initiate a basic research investigation in new materials and lubricants for future applications.

3. RESEARCH DISCUSSION

The following outline discusses this program in greater detail.

3.1. Bearings:

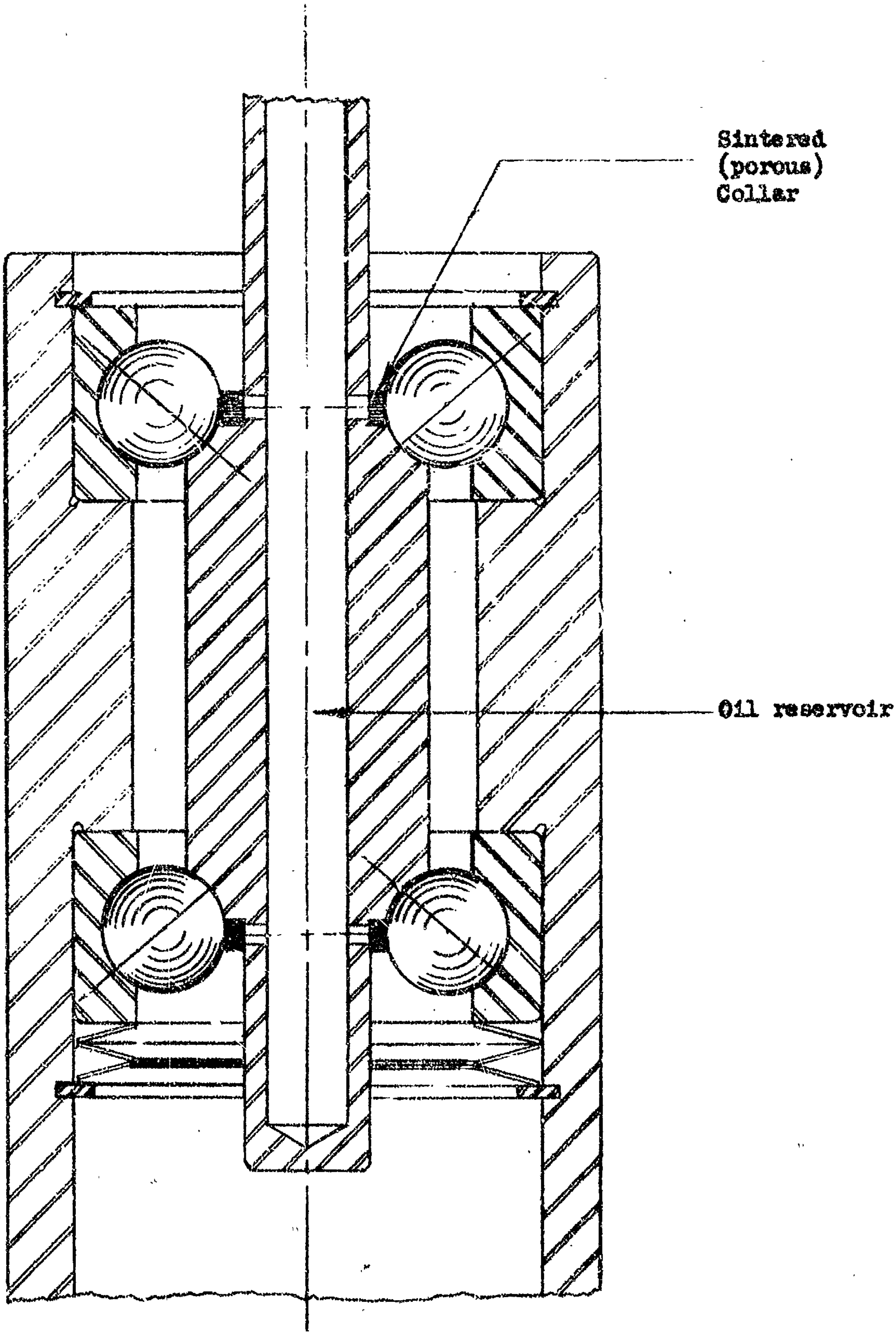
It is recommended that research and development efforts on bearings be conducted simultaneously in three areas: to extend the range of oil lubrication, to develop materials and lubricants for space environment, to develop mechanical designs to accommodate specific problems.

3.1.1 Oil and Grease Lubrication:

To extend the range of oil lubrication requires the development of suitable vacuum seals. This is discussed separately elsewhere in this report, however, possibilities for carrying increased quantities of lubricants also present themselves.

Present production bearings are supplied with retainers impregnated with approximately 5% of lubricant by weight. Recent research effort at MRC has succeeded in raising this level to approximately 8-10% by weight. Further work in this area is warranted. For any material, an optimum level of impregnation is expected to exist, beyond which, life would reduce due to increased flow rate. However, investigation should include the use of new materials capable of increased impregnation and investigation of techniques for preparing these materials to permit increased impregnation. These studies should be expanded to include the use of oils more favorable to vacuum environment (low vapor pressure oils).

Work has also been underway to utilize large quantities of oil carried in chambers integral with the shaft. This oil feeds through the shaft wall to porous material imbedded in the inner race adjacent to the ball path. This is illustrated in Figure 1.



INTEGRAL SHAFT LUBRICATION

Figure 1

Work by Aeronautical Systems Division (WADD) and Mr. R. J. Matt of Thompson-Ramo-Wooldridge (Ref. 68) indicates that a liquid lubrication system is feasible for operation in a zero gravity environment.

Work has also been conducted to extend the duration of grease lubrication in high temperature applications. This work should be continued, at lower temperatures, directed to low pressure environment. Lubrication with grease for extended life requires the provision of increased grease space adjacent to the bearing, provision for feeding the grease into the bearing as required, and a grease which leaves no residue detrimental to bearing performance. High temperature work has shown progress in all three areas. Figure 2 shows an example of the type of grease chamber which has been investigated. The contour of parts A and B are constructed in a manner to gently urge the grease toward the bearings.

3.1.2 Materials Development:

Material development for antifriction bearings should include the following:

a. Evaluation of load and speed capability of Pyroceram bearings.

b. Performance evaluation of ring and ball materials in vacuum in bearing assemblies.

c. Dry film -

    1. Evaluation of performance in vacuum in bearing assemblies.
    2. Evaluation of retainer-dry film combination.

d. Investigation of the basic theory of dry film lubrication including the development of new dry film materials.

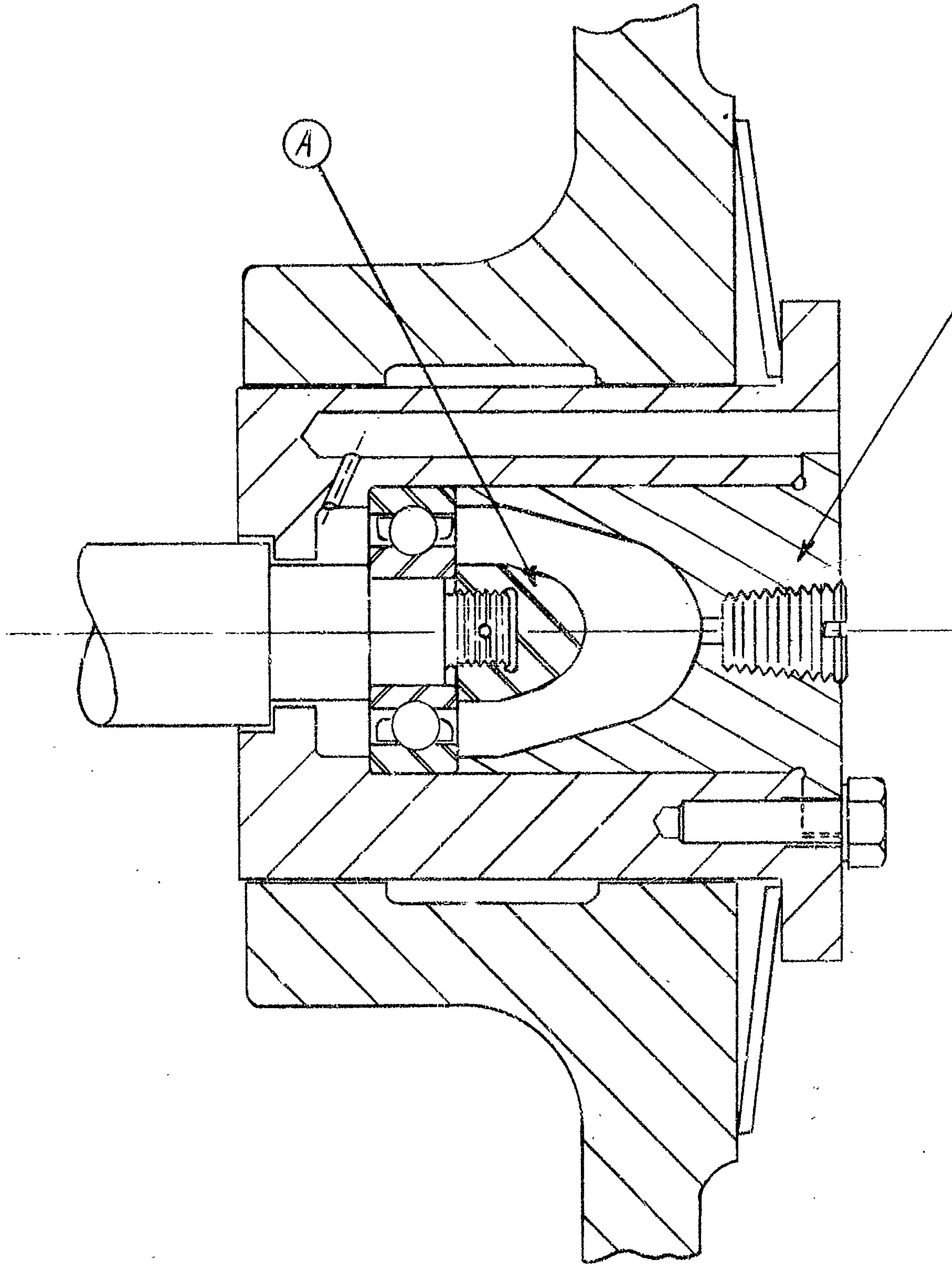
e. Development of new ring and ball materials.

Considerable effort has been expended in the development of Pyroceram as a bearing material. Manufacturing Research has resulted in techniques which have permitted hundreds of bearings to be fabricated of this material in production. Sufficient testing has been conducted to establish mechanical design requirements and demonstrate reasonable load capacity at speeds to 10,000 RPM. This is an excellent candidate as a ring material for vacuum exposure. An immediate program should be initiated to revaluate the load and speed characteristics of Pyroceram bearings and the wear and friction performance in vacuum.

The need for basic data concerning wear and friction of bearing materials in a vacuum environment is demonstrated by the inclusion of the friction experiment in Ranger 1 and 2 (Refs. 6, 8). Many of the data which have been generated in laboratory experiments are subject to considerable question due to failure to adequately duplicate the space environment (Ref. 58). Sliding friction experiments, while desirable, are not sufficient. The suitability of material combinations must be finally substantiated by full-scale bearing tests.

Research on dry film should include wear and friction analysis in full-scale bearings. Useful life is more apt to be determined by limiting friction and wear than the classical fatigue failure associated with fluid lubricated bearings. Experimentation to date has been primarily concerned with developing data for specific applications, at extreme high temperatures, and in air or gaseous environment. General information, comparable to conventional catalog information for oil lubricated bearings, should be developed for the designer's use in space environment.

Several classes of dry film lubricant materials should be considered. These may be classed as organic chelating materials, soft metal films, and inorganic compounds. Examples of these include metal free phthalocyanine, silver and the soft metal oxides halides, sulfides and tellurides (Refs. 14, 80).



EXPERIMENTAL GREASE CHAMBER

Figure 2

Effort should be directed to the application technique for dry films. Experience at the U. S. Naval Air Material Center has shown that such variables as particle size and base material surface preparation play a large role in successful lubrication with MoS_2 type dry film. The need for reliable, production application techniques must not be overlooked.

Use of dry film also influences detail bearing geometry. Experience has shown that radial clearance may have to be larger than is customary in order to accommodate wear debris and dry film particles scuffed from the surface during rotation (Ref. 110).

There is also difference of opinion as to the best location for application of dry film within the bearing. Our own preference has been to lubricate the retainer only, while others prefer to lubricate both retainers and race surfaces. Insufficient information is available to compare these different approaches.

The combination of dry film and base materials has been shown to be extremely important (Ref. 108). An investigation of the basic theory of lubrication with dry film is warranted, especially encompassing the influence of space environment. A more thorough understanding is needed of the role which the base material plays in the ability of the dry film to function as a lubricant. This will lead to the need for the development of new ring and ball materials, and new dry film materials specifically tailored to each other.

There is also much promise in attempts to utilize dry film lubricant materials to fabricate complete retainers.

Various ceramics and cermets have shown characteristics in friction tests in inert environments which indicate that complex oxide films may be formed at the interface of the materials to reduce friction and wear. Candidate materials for study of these phenomena are: Silicon Carbide Ceramic, Titanium Carbide Cermets (TiC with nickel binder or a binary nickel-molybdenum binder), and metal oxide ceramics such as aluminum, magnesium, and beryllium oxides. Refractory metals, such a molybdenum and chromium are of interest as cermet binders. The cermet Al_2O_3-Cr-Mo has exhibit better friction and wear performance in helium than in air (Ref. 85).

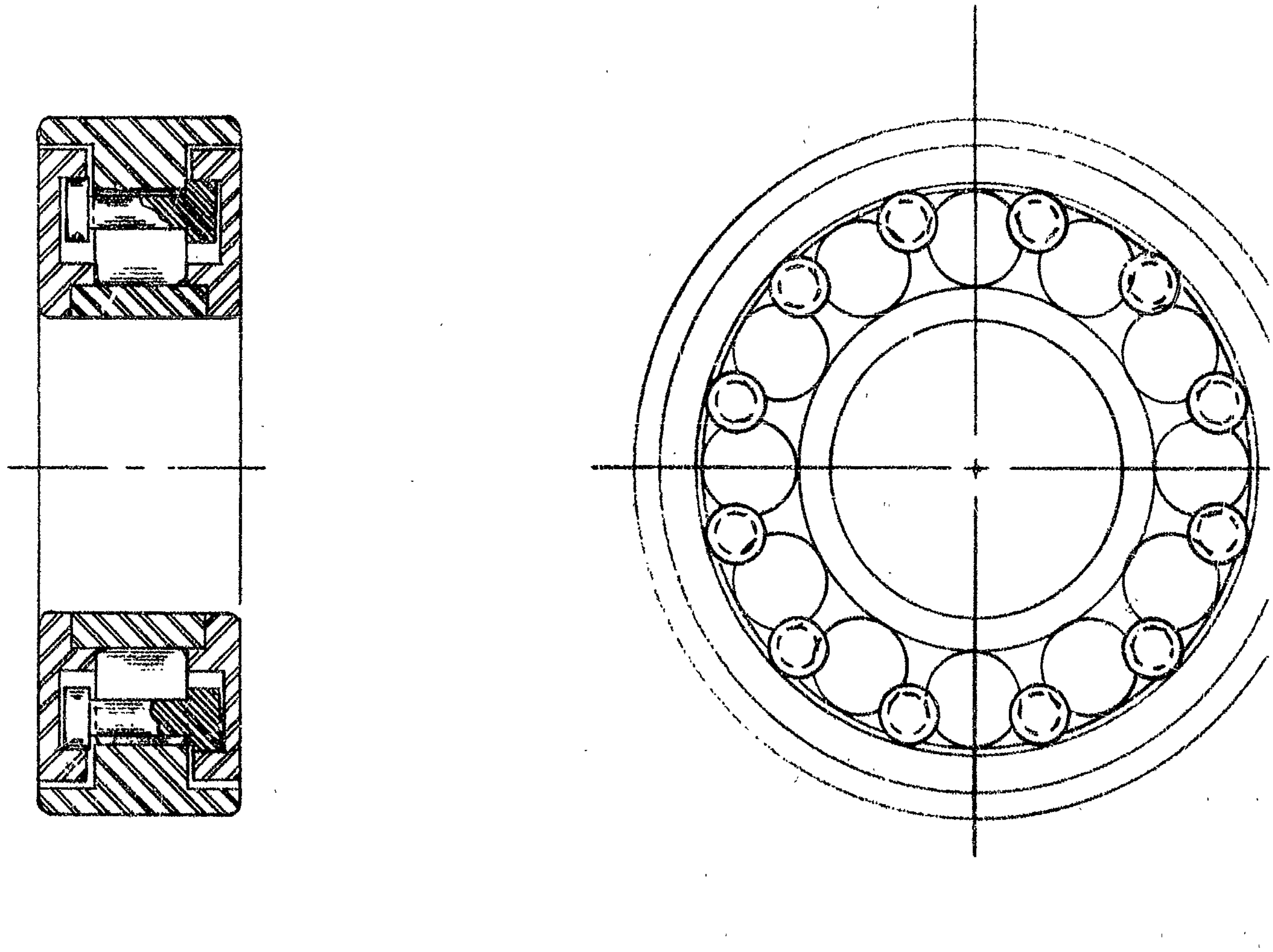
3.1.2 Mechanical Design of Antifriction Bearings:

While simple in its mechanical concept, a satisfactory bearing requires precise con trol of many geometric variables. When one thinks in terms of 100% reliability for periods of two years or more in the hostile environment of space, the effect of the details take on added importance.

The effect of race curvatures, contact angle, preload, etc. on reliability, limitin speed, wear and friction must be considered. The effect of thrust versus radial loading and combined thrust and radial loading is particularly significant.

Investigations are required of the extent to which the mechanical design of the bea ing needs to be compromised to permit self-lubricating bearings. To provide a sati factory bearing from Pyroceram material, it has been necessary to reduce the ball complement and to redesign the retainer to permit sufficient shoulder area for this brittle material. The retainer is a logical lubricant carrier in any self-lubricating bearing. If the ball complement is reduced, more flexibility in the retainer design, such as the use of lubricating buttons, is permitted. Other suggestions which require design studies include the use of alternating balls of self-lubricatir material, lubricating ring adjacent to the ball path, and the structural support or reinforcement of self-lubricating retainer material.

The bearing shown in Figure 3 represents a radical departure from conventional beari design in an attempt to minimize the areas of sliding friction. Theoretically, all contact between elements of this bearing occurs with pure rolling motion. This bear



ROLLING ELEMENT BEARING

Figure 3

was originally conceived at NACA (later NASA) for use under conditions of marginal fluid lubrication. Conventional ball bearings proved to be satisfactory in that experimental program and further work on this configuration was dropped. Recently, experimental bearings of this type have been fabricated and tested at MRC. Operation, dry, at room ambient conditions has demonstrated that sliding friction is reduced sufficiently to permit the bearing to operate for extended periods at stabilized temperatures comparable to oil lubricated bearings. This work was done with a roller bearing configuration. Fabrication of a similar ball bearing has been completed and preliminary tests are underway. A thorough evaluation of load, speed and life characteristics of this design is warranted.

Numerous other mechanical designs which have received attention, such as gas bearings, and electromagnetic bearings have not been included within the scope of this report.

3.1.4 Plane Bearings:

The most serious problem in the application of plastic materials to plane bearing operation is the limiting speed imposed by their low heat conduction. While bonded plastic and plastic impregnated metal offer promise of higher speed, satisfactory design data is not available. Limiting speed versus PV should be evaluated for these bearing types, together with combinations involving self-lubricating fillers such as molybdenum disulfide. The effect of vacuum and radiation on wear and friction likewise requires experimental investigation.

While inorganic materials such as sapphire, do not suffer from these temperature limitations, their performance in vacuum has not been sufficiently evaluated. Friction coefficients tend to be relatively high, indicating a requirement for experimentation with lubricant coatings.

3.2 Gear Development:

3.2.1 Oil and Grease Lubrication:

As in the case of bearings, it is desirable to extend the range of oil lubrication for gears as far as possible. Again much depends on the ability to develop a satisfactory seal. In the case of gears, however, the problem is further compounded by the problem of retaining and circulating oil in a zero gravity field.

While present practices avoid fluid lubrication in gear boxes, it does not seem feasible to continue to avoid this problem if a satisfactory seal can be developed. Heat dissipation may become a major problem in highly loaded gearing. Friction is known to be critical because of low power availability and some beam failures of gear teeth have been experienced even though most gearing is relatively lightly loaded. Since gears must run over such a large range of speeds, reduction ratios and operating cycles, it is reasonable to assume that problems of sufficient magnitude will be encountered to require oil lubrication.

It should be feasible to develop methods of lubrication similar to that now used in bearings with porous retainers. It should also be feasible to design grease feeding systems suitable for gear chamber use. A program in this direction certainly appears warranted.

3.2.2 Materials Development:

There is no known data conerning the operation of gears in vacuum. The performance of dry film lubricated gears in inert gas environments is very promising. There is a need, however, to develop wear and friction data as a function of tooth load and speed in the space vacuum environment.

The means of storing and providing the dry film as a lubricant warrants investigatic One possibility is the use of idlers, meshing with the gears but not part of the powe train, as lubricant carriers.

Mechanical design development will be required to provide satisfactory transfer of lubricant from such an idler. Means must be provided to load the idler with its mating gear with sufficient tooth scrubbing action. A worm type idler may be advantageous.

Basic studies of base gear material and dry film combinations in the field of antifriction bearings will be applicable to gear development. Care should be exercised that these programs take a direction which will provide suitable information to the gear designers.

Self-lubricating materials should also find application in gear trains. There is, however, a need for basic performance analysis of strength, wear, friction, speed and load limitation in vacuum and radiation environment.

The known strength limitations of plastic gearing may severely limit their application. However, plastic self-lubricating materials may still find application in gear trains used as impregnation in gear metals or as plastic idler gears used as lubricant carriers adjacent to the power train.

Promising material development is now being carried out at Western Gear Corporation, Lynwood, California. Laminated gears consisting of a soft core for strength and a hard coat for wear and friction reduction are being developed using plasma spray techniques. The need for basic material development overlaps the area of bearings and points up the need for coordinated effort for these two fields.

3.2.3 Mechanical Design:

The continuing need for extremely high reduction ratios makes a suitable worm gear most desirable and efforts in mechanical design innovations should include this problem. One concept, to reduce the inherent high sliding friction, is that of a ball screw in which the shaft is in the form of a circular arc. Such devices are suitable for partial rotation and have been used in automobile steering mechanisms. It is conceivable that this concept could be extended to complete rotation by providing a means to externally support the circular, threaded shaft.

Consideration must be given to the effect of self-lubrication on the mechanical design of gearing. Experimental effort is needed to resolve design questions on manufacturing precision, tooth pitch, reduction ratios per stage, tooth form, beam strength, and so forth.

Mechanisms involving parallel shaft centerlines afford an opportunity for the application of design ingenuity to minimize sliding contact:

a. Roller Chain Drives - In this concept the sliding motion of a conventional gear tooth against its mate is replaced by rotation of a bushing on a pin. While sliding still exists in the bushing to pin contact, the configuration is far better suited for the retention of lubricant and reduction in contact stress.

b. Rocking Link Chain - In this configu tion the rotational motion of the bushing on the pin of a roller chain is replaced by a rocking contact involving practically pure rolling. This design is conventionally manufactured in a different chain type known as an inverted tooth type chain. A small sliding component is involved in the contact of the sprocket to chain tooth.

Other mechanical innovations which suggest themselves are:

a. Geneva Mechanisms - In this design concept torque is transmitted by the pressure of an antifriction cam roller against a track. While these devices usually involve intermittent motion or a dwell in the velocity curve, they can be designed to closely approximate continuous rotation. This concept may be used with parallel as well as cross axes.

b. Loose Tooth Gears - Rocking couples find frequent use to transmit motion from one shaft to another in pure rolling contact (Ref. 73). In this design configuration, forces are transmitted from the teeth of one gear to another through a third member interposed between them which is free to align itself radially during contact.

It is anticipated that ball screws will find application in future space vehicles. Effort is required to determine the design criteria of ball screws utilizing non-fluid lubrication.

3.3 Seals:

As emphasized in the report on hyperenvironment simulation by T. M. McCoy of Northrup Corporation (Ref. 67), at relatively low pressures the physical concept and significance of the term pressure takes on an added meaning from that of the more common place surface environment. This must be recognized in seal development. Seal development should be approached from the scientific viewpoint, utilizing the molecular flow concepts which take into account the significance of molecular mean free paths to the pressure properties of a gas. The feasibility of this approach has been demonstrated in Tiros II (Ref. 66).

Work on static seals is encouraging (Ref. 109) but demonstrates the magnitude of the problem.

Work on combination seals involving clearance seals, rubbing seals, labyrinths, and porous plugs is warranted.

Effort should be directed at developing suitable face type seals. While material and lubrication problems are similar to plane bearings, they involve the further complication of minimizing escape path distance between seal load supporting asperities. Effort is needed to evaluate performance versus pressure drop including evaluation of optimum face pressures in the light of wear rate and power absorption for dry film lubricants and various self-lubricating materials.

For oscillating applications, such as the leg joints of walking vehicles, boots offer promise. Development of boot materials is required. Optimum pressure levels with respect to leakage characteristics and the effect of internal pressure and stress on flecture life must be determined. The work of Mechtronics Corporation (Ref. 43) in the area of long life, precision metal bellows is significant.

3.4 Research on Electrical Contacts:

Dr. Wilfred E. Campbell (Appendix A) emphasizes that research should begin with a development of experimental techniques to study the problems of electrical contacts in space environment. There is a basic lack of knowledge of how current is conducted through electrical contacts. Thus, the space environment problem is a specific case of a more general problem which is not fully understood. Before significant progress can be made, for the long range future, there is need to establish an electrical contact model which will probably be very complex in nature. Effort should serve to explain these phenomena:

A. The mechanism of current conduction through contact.

B. Erosion of contact and asymmetry.

C. The nature of glow discharge and its effect on erosion of contact.

D. The effect of current, pressure, and the type of contact motion on the wear rate of contact materials.

E. The nature of product of electrolysis in vacuum.

F. An explanation for fluctuations in contact resistance (noise).

This will require a major research effort.

Immediate effort is required on the development of materials for contacts in vacuum. This should include the effect of current on friction, wear, and evaporation rate of materials in vacuum. For the materials which are of particular interest due to their electrical conducting properties, an analysis of molecular adhesion, chemisorption, shear strength and eutectic forming properties in vacuum should be investigated.

There is a need for development of lubricant to minimize mechanical wear and a need for design studies to minimize wear and friction. Methods need to be found to replenish lubricating films on contact surfaces. The dust collecting tendencies of various lubricants should be studied, as well as the effect of vacuum on brown polymer formation associated with contacts.

Section F

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